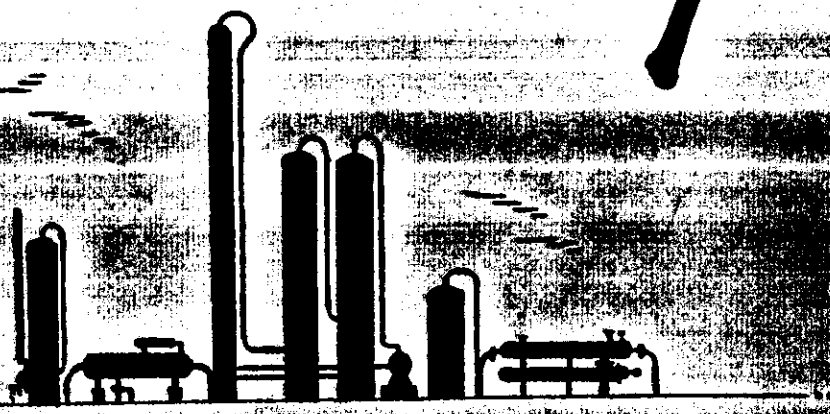
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ELEMENTS OF

field processing



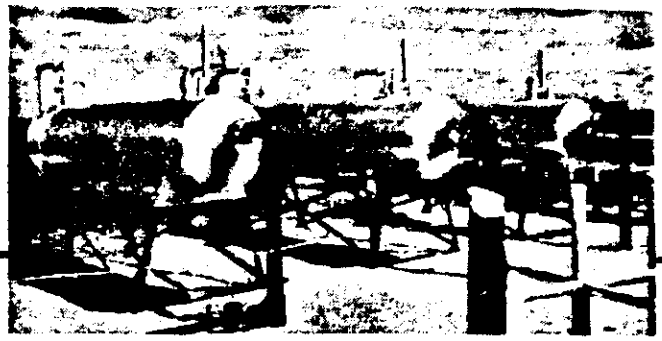
PART 1

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ELEMENTS OF FIELD PROCESSING . . . Part 1 of a Series

Know Your Separators



. . . They are the key to field processing

If separators are properly sized and operated, the loss of oil and rich vapors will be held to a minimum. More oil and higher gravity mean more money for the producer. The series that starts with this article is intended to show the operator how to choose and operate his separators and other field-processing equipment for maximum profit. The next few articles will show the effect of pressure and temperature on separation, application of stage separation, and how to get top efficiency from other operations.

by John M. Campbell

SEPARATION of oil and gas is the most critical of the field-processing operations although it is still the most common. As producing pressures have risen and lighter condensates are produced, efficient separation has become more critical than ever. Moreover, some of the new concepts have been applied to advantage on old leases producing oil at moderate pressures.

As gas-transmission lines raise their standards, separation becomes a part of the over-all field processing necessary properly to condition the gas. Consumer specifications are becoming more rigorous; it is only a matter of time, probably, until most gas contracts will specify dust and liquid content in addition to the present requirements.

Separation requirements . . . A "complete" separator must have the following:

1. A primary separation section to remove the bulk of the liquid from the gas.

2. Sufficient liquid capacity to handle surges of liquid from the line.

3. Sufficient length or height to allow the small droplets to settle out by gravity, to prevent undue entrainment.

4. A means of reducing turbulence in the main body of the separator so that proper settling may take place.

5. A spray condenser to capture entrained droplets or those too small to settle by gravity.

6. Proper back-pressure and liquid-level controls.

All separation may be divided into two classes—scrubbers and separators. This classification is loose, and is only to distinguish that equipment having all the above features from that which does not. In this nomenclature, a scrubber is any device designed for separation of liquid from gas that does not have features (2) and (3), and possibly (6). It is a vessel, then, designed to handle relatively small quantities of liquid with no large degree of surging.

Choice of the separation device obviously hinges on economics. By virtue of its smaller size and simplicity, a scrubber is cheaper in initial cost and easier to install. Use of scrubbers should be carefully examined to see what ultimate effects liquid carryover might have on operations downstream. Such an examination often reveals that the slight additional cost of a separator is indeed very cheap insurance.

It is poor practice to use a scrubber as the primary separation means at the well. Production of liquid is often unsteady and there is a definite need for liquid surge capacity. Even dry-gas wells will often start producing liquid, including salt water, during their later life. Consequently, scrubbers are recommended only for:

1. Secondary separation to remove carryover fluids from process equipment such as absorbers and liquid dust scrubbers.

2. Gas-line separation downstream from a separator and where flow lines are not long.

3. Miscellaneous separation where the gas-liquid ratio is extremely large.

Types of separators . . . The separators of various manufacturers differ slightly in details, but all may be subdivided into (1) vertical, (2) horizontal, and (3) spherical. The spherical type is a fairly recent addition, having been offered by some manufacturers for competitive reasons. Although it embodies all features of conventional separators, it has limited liquid surge capacity and separation space. According to the nomenclature used here, it would fall on the line between a scrubber and a separator. Its use appears limited to small leases, operating at moderate pressure, where initial investment is considered important.

Table 1 summarizes the advantages and disadvantages of each type. Operating data on the spherical type are limited so a final evaluation is difficult.

Table 1 shows that the vertical-type separator is the most versatile but the most expensive. Consequently, its use must be justified by actual operating requirements or by that intangible "operating insurance." On the average lease, the extra cost of a vertical separator, particularly in the smaller sizes,

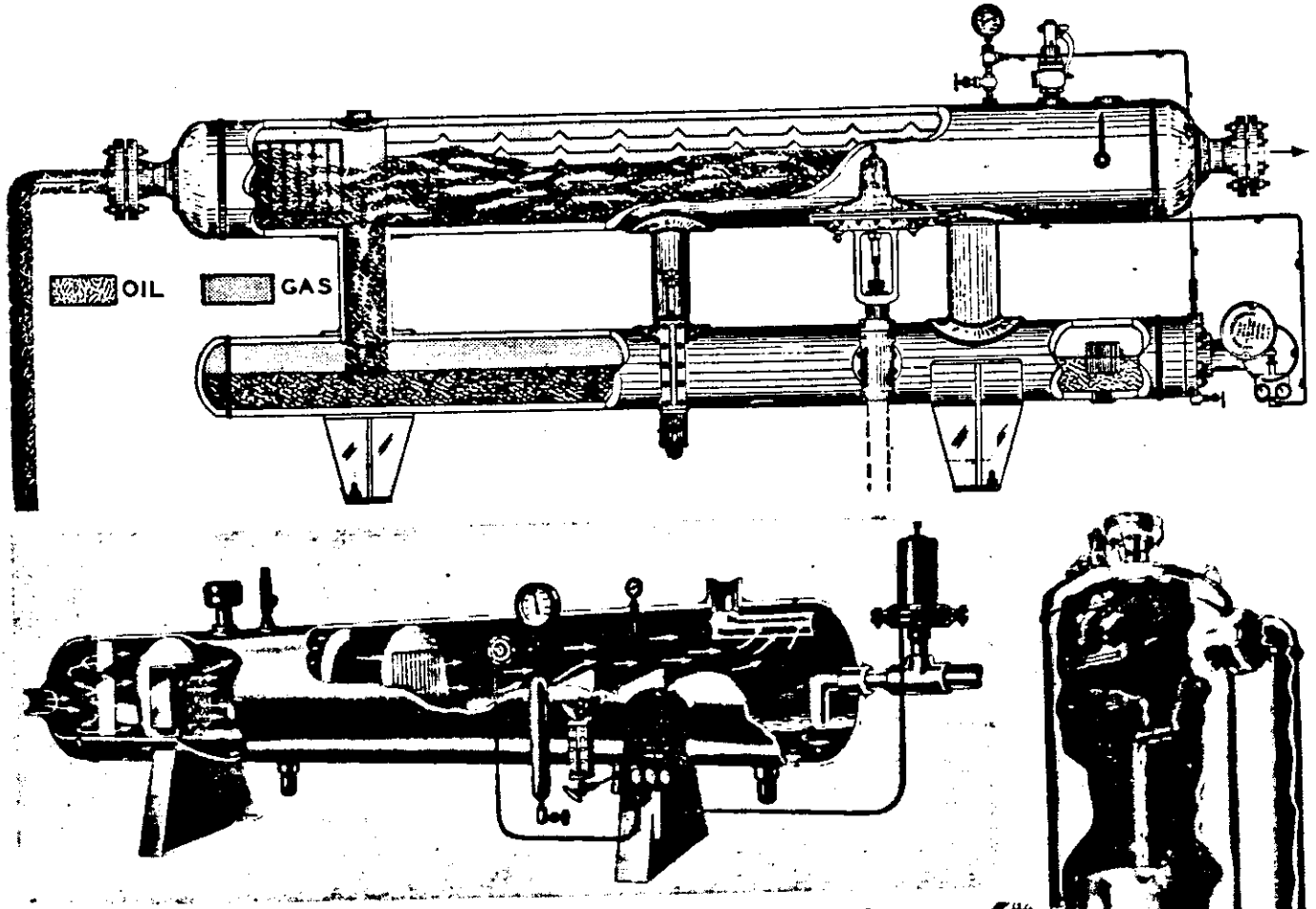


Fig. 1—A double-barrel horizontal separator above and a single-barrel below. The main advantages of this type are low cost and ease of mounting and piping.

is negligible when compared to the total well cost. Where the gas-oil ratio is low, the vertical type is always recommended.

Separator components . . . Figs. 1 and 2 are typical cutaway views showing the various separator parts.

Primary separation is normally accomplished with the aid of centrifugal force. Either a tangential inlet or internal piping is used to cause a spinning motion of the incoming gas. The latter is preferred by most since it permits in-line piping. With this approach, the large liquid droplets are thrown against the walls, which limits the upward flow and reduces entrainment.

All standard separators have capacities based on nonsurging of fluids. However, experience has dictated that such surge capacity be built in. There is no way of determining the necessary surge capacity as the degree of surging can never be known in advance. Consequently, the amount of capacity has to be a compromise between cost and the average process requirements.

TABLE 1—SUMMARY OF SEPARATOR OPERATING CHARACTERISTICS

Type	Advantages	Disadvantages
VERTICAL	Liquid level control not quite so critical. Will handle larger quantities of sand and mud without foiling. Easier to clean out. Has greater liquid surge capacity as usually manufactured. Less tendency for revaporization of liquid.	Most expensive. On skid assemblies more difficult to ship. Takes larger diameter for a given gas capacity.
HORIZONTAL	Cheaper than vertical. Easier to ship on skid assemblies. Easier to pipe up. For a given gas capacity diameter smaller than vertical.	Liquid level control more critical than with vertical type. Difficult to clean of mud, sand and paraffin. More area available for settling when two liquid phases are present.
SPHERICAL	Cheaper than either horizontal or vertical. Compact.	Has limited separation space and liquid surge capacity. Liquid level control critical.

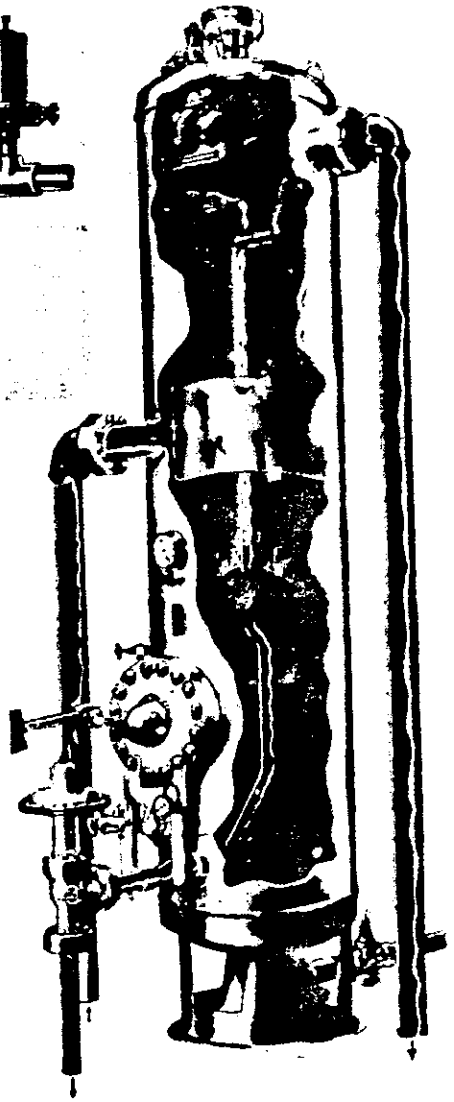


Fig. 2—A vertical separator. Its advantages are ease of cleaning, lack of revaporization and ability to handle liquid surges and large volume of sand.

Standard-size separators, produced by reputable manufacturers, have very similar dimensions; and all have, in effect, incorporated an arbitrary amount of surge capacity. In many cases, particularly with the vertical type, shorter

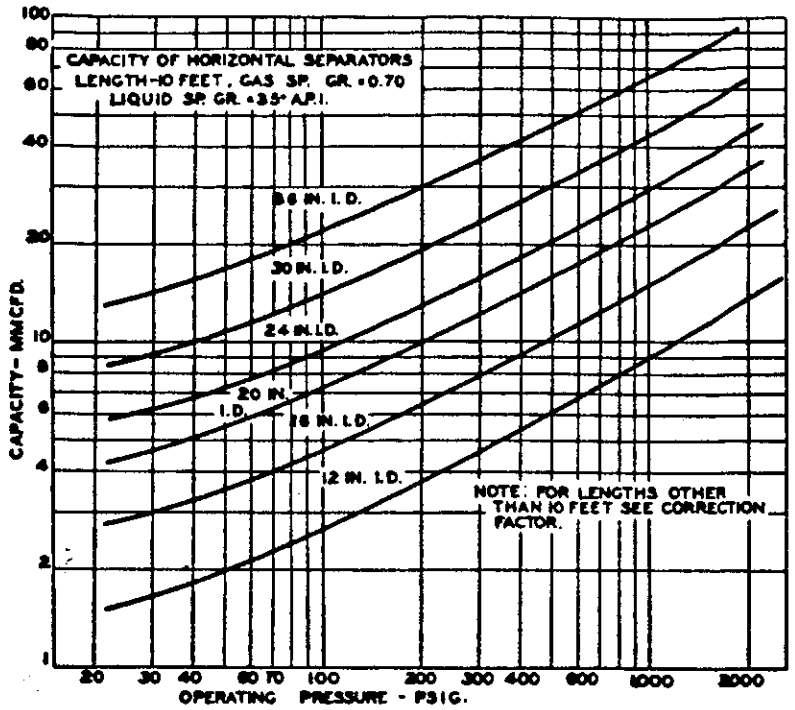


Fig. 3—These curves are based on a 10-ft. length, but at higher pressures a longer separator may be cheaper.

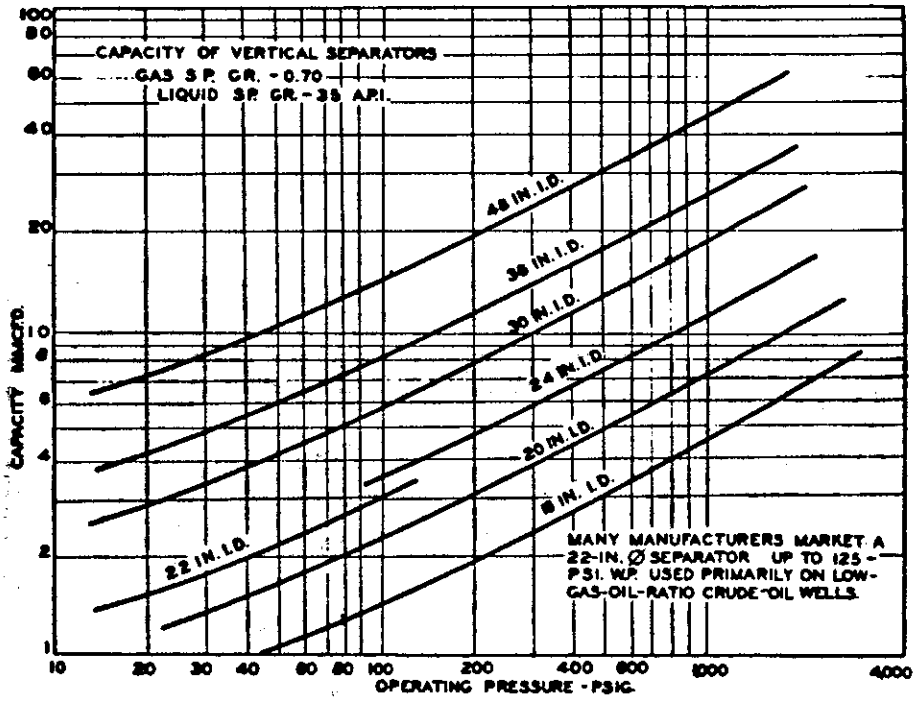


Fig. 4—A difference of 2 ft. between oil inlet and outlet is considered in these curves. With less, there may be undue entrainment.

separators have proved entirely satisfactory.

The length of a horizontal separator has a greater effect on capacity than the height of a vertical type. In the horizontal vessel, the path of any droplet ideally has a trajectory similar to that of a shell from a gun. Therefore, the length necessary depends on:

1. Droplet size.
2. Gas velocity.
3. Droplet density.
4. Vessel diameter.
5. Degree of turbulence.

In the ideal case, turbulence may be neglected and effect of gravity assumed constant. Droplet velocity affects length for as it increases, the droplets will travel farther before settling out. Although corrections have to be made for the nonideal conditions that exist, this analysis shows that the capacity of horizontal separator of a given diameter may be increased by increasing length.

Application of the same principles to vertical separators shows slightly different results. There, the droplet velocity is working against gravity, which makes separation more difficult. In order to keep separator height reasonable, velocities must be smaller. Sufficient length is needed only for the velocity to become zero and for the droplet to start falling. Increasing the length above this point accomplishes little good.

If too much liquid is carried into the separation section, the falling of the small droplets is hindered and entrainment becomes more prevalent. This is one reason internal design is critical. Theory does not give the designer sufficient tools, so design is best accomplished by actual test.

The spray condenser is designed to trap those droplets that have for any reason been carried beyond the separation section proper. This subject will be explored more thoroughly at a later date.

Single and double barrel . . . The horizontal separator in Fig. 1 is known as the double-barrel type and consists of an upper separation section and a lower liquid chamber. The single-barrel type combines both into a single chamber. A difference of opinion exists as to which is the best as each has its inherent advantages. Either should work satisfactorily if properly designed and used within its limitations. In many cases, initial cost is the prime consideration. Table 2 summarizes the advantages advanced for each type.

To prevent undue liquid turbulence and reentrainment of liquid, most manufacturers provide some sort of baffle plate above the liquid level on the single-barrel type. To have a gas capacity equivalent to that of a double-barrel type, the diameter must be larger to compensate for the volume occupied by the liquid.

Sizing of Separators

Since theory does not take some important factors into account, published separator capacities have been obtained by test and the application of rather arbitrary standards.

Gas capacity . . . For many years designers have used variations of the empirical formula:

$$Q = 67{,}824\ CD^2\ (1/Z)\ (P/P_b) \times [(\rho_L - \rho_G)/\rho_G]^{1/2} \qquad (1)$$

to determine the gas capacity of separators, where:

Q = gas capacity, standard cu. ft. per day
C = proportionality constant
D = internal diameter of separator, ft.

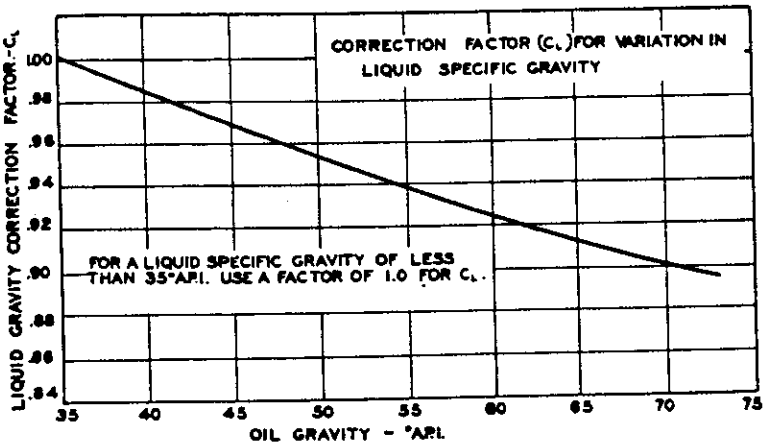


Fig. 5—The C_L factor corrects for oil gravity.

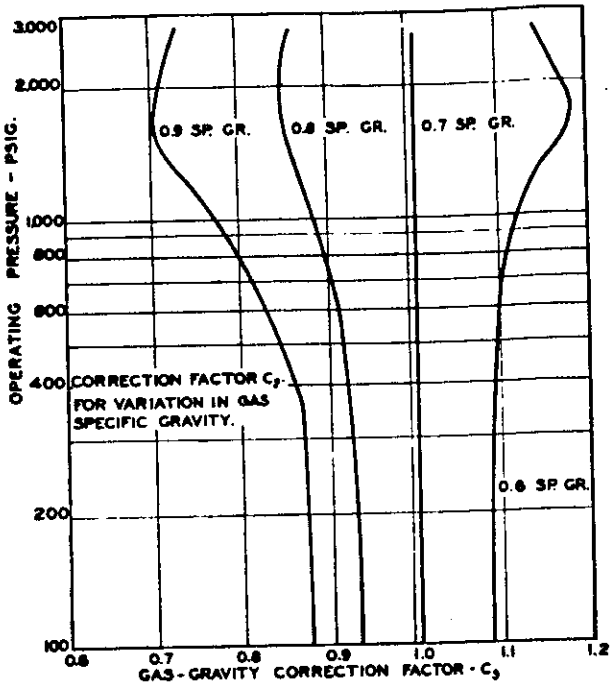


Fig. 6—If the gas gravity is not 0.70, the C_G factor from this chart will help to find capacity of a separator.

Z = compressibility factor, dimensionless

P = operating pressure, psia.

P_b = pressure base, psia.

ρ_L = density of liquid, lb./ft.^{3}

ρ_g = density of gas, lb./ft.^{3}

Use of Equation 1 in a vertical separator stems from analogy with allowable vapor velocity in fractionation columns and absorbers. Consequently, a value of $C = 0.16$ to 0.18 is often used. For horizontal separators the same equation has been used with $C = 0.40$ to 0.42.

In absence of detailed capacity charts, Equation 1 may be used for estimating separator size requirements. Many calculators assume ideal gases ($Z = 1.0$) for sizing calculations, particularly at pressures below 1,000 psig. An additional temperature correction could be shown; but in most separation processes, it is not significant except for its effect on liquid and gas gravities.

At least one reputable manufacturer bases sizing curves on the following conditions:

1. Smallest liquid particles separable are spheres 10 μ (0.0004 in.) or larger in diameter.
2. The specific gravity of the liquid particles in the gas stream is 45° A.P.I. or less.
3. No liquid foaming is present and there is no undue surging.
4. The oil is at a temperature above its cloud point.
5. The gas is above the hydrate forming temperature.
6. The gas has a specific gravity of 0.7 and the separated oil is 35° A.P.I.

Using these assumptions, the entrainment of liquid through the separator is guaranteed not to exceed 0.1 gal. per 1,000 standard cubic feet.

A means of estimating separator gas capacity is provided by Figs. 3-6. Figs. 3 and 4 are actually a conservative composite of various manufacturers' curves and check Equation 1 rather closely. In these figures, internal diameter is used as a parameter. Many manufacturers base separator sizes on the outside diameter so that a correction for wall thickness would be necessary. This is easily done for from Equation 1 it may be seen that Q is proportional to D^2 for a given well stream, at given separation conditions. Consequently,

$$Q_1 = Q_2 (D_1^2/D_2^2) \qquad (2)$$

where

Q_1 = capacity desired for a separator with i.d. D_1

Q_2 = known capacity for a separator with i.d. D_2

The vertical separator capacities shown in Fig. 4 are based on the use of a tangential inlet (or the equivalent) and at least 2 ft. difference in height between the oil outlet and inlet. Many designers feel that a minimum of 2 ft. is required to prevent undue entrainment as a result of turbulent effects.

The horizontal separator capacities shown in Fig. 3 are based on separators 10 ft. long with the equivalent diameter shown being entirely available for gas flow. For single-barrel separators, that portion occupied by liquid should be subtracted from the total area to find an equivalent diameter.

At high pressure it is less expensive to build longer separators because of the saving in the thickness of shell plate used, which in turn reduces fabrication costs. Published data show that the following equation may be used for estimating the capacity at any length once the capacity for some other length is known:

$$Q_1 = Q_2 (L_1/L_2)^{0.56}$$

where

Q_1 = capacity of separator of length L_1

Q_2 = capacity of separator of length L_2, with same diameter as L_1

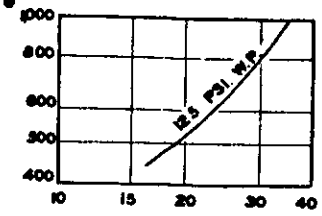
When using Fig. 3, Q_2 would equal the capacity read from the graph and $L_2 = 10$ ft.

For those cases where the gas and liquid gravities are different from 0.70 and 35° A.P.I. respectively, it is necessary to use Figs. 5 and 6. Fig. 5 is a plot of correction factor versus oil gravity in °A.P.I., while Fig. 6 gives a corresponding correction factor for gas.

TABLE 2—SUMMARY OF ADVANTAGES CLAIMED FOR SINGLE AND DOUBLE BARREL HORIZONTAL SEPARATORS

Single barrel:
1. Lower first cost.
2. Easier to insulate for cold weather operation.
3. Gas and liquid remain in contact, keeping the liquid warmer and minimizing freezing and paraffin deposition.

Double barrel:
1. Under surging conditions larger capacity with less upset by diverting bulk of liquid to lower chamber at inlet. The separation section is semi-isolated from the effect of such surges.
2. Better separation of solution gas is obtained in the quiescent lower chamber.
3. Better separation when gas and liquid have similar densities, as at high pressure.
4. Less chance for reentrainment.
5. More stable liquid level control as the float is located away from the turbulent gas-separation section. Level control is not quite so critical.



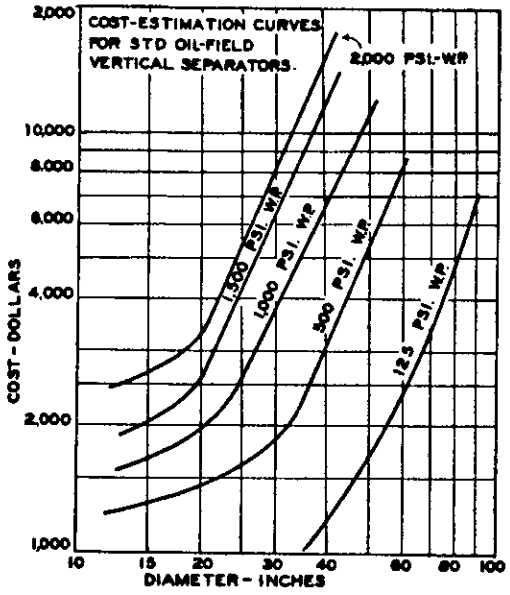


Fig. 7—A fair estimate of first cost of vertical oil-field separators can be made from these curves.

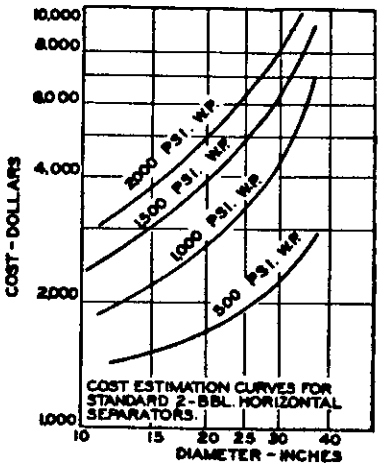


Fig. 8—If it were not for the different cost of fittings, all these curves would meet at some small diameter.

The total capacity of a separator is then found from the equation

$$Q_T = (Q)\ (C_g)\ (C_L) \qquad (4)$$

where

Q_T = capacity at actual conditions
Q = capacity read from Fig. 3 or 4
C_g = gas correction factor from Fig. 6
C_L = liquid correction factor from Fig. 5

Example: Determine the gas capacity of:

(a) 30-in. i.d. vertical separator
(b) 30-in. i.d. horizontal separator, 10 ft. long
(c) 30-in. i.d. horizontal separator, 20 ft. long

when processing a well stream containing 0.80-sp. gr. gas and 50° A.P.I. liquid, separating at 800 psig.

Solution:

(a) From Fig. 4 gas capacity = 16.6 M.M.c.f.d.
From Fig. 5 $C_L = 0.952$
From Fig. 6 $C_g = 0.90$
$Q_T = (16.6)\ (0.952)\ (0.90) =$ **14.2 M.M.c.f.d.** (from Equation 4)

(b) From Fig. 3 gas capacity = 39.0 M.M.c.f.d.
From Fig. 5 $C_L = 0.952$
From Fig. 6 $C_g = 0.90$
$Q_T = (39.0)\ (0.952)\ (0.90) =$ **33.4 M.M.c.f.d.**

(c) $Q_2 = 33.4$ M.M.c.f.d. and $L_2 =$ 10 ft.
$L_1 = 20$ ft.

From Equation 3

$Q_1 = (33.4)\ (20/10)^{0.50} = (33.4) \times (1.47) =$ **49.1 M.M.c.f.d.**

Liquid capacity . . . Liquid capacity is limited by two factors—liquid capacity of the downcomers and other liquid handling portions of the vessel, and/or capacity of the liquid-level control valve. Since there are infinite combinations of these, no general correlation of liquid capacity is feasible.

As the pressure increases, the capacity will do so correspondingly because more driving force is available to force the liquid through the vessel and across the valve. However, no effective change in capacity is usually specified after the pressure exceeds 1,000 psig.

One method of increasing separator capacity is the use of a larger stem and seat in the dump valve. This should be done with caution, though, for many control complications may arise. This aspect will be discussed in detail in a future installment.

A large majority of those separations, where the liquid capacity is governing, are on low-pressure crude-oil wlls. As a guide the following capacities are given by one manufacturer for 125 psig. working pressure vertical separators.

Diameter (in.):	Dump valve size (in.)	Capacity bbl./day
22	2	600
30	4	1,850
36	4	2,760
48	4	5,500
60	6	8,600

The capacities shown are also based on the liquid properties previously assumed.

Separator costs . . . Figs. 7 and 8 give curves for estimating the initial cost of standard oil-field separators. Such curves are never entirely accurate because the variation of fitting and control cost has a marked effect on total cost, particularly with small-diameter vessels. If it were not for this, all cost curves should converge to a single point at some small diameter because a minimum wall-thickness vessel would suffice for all pressures.

The curves generally give investment costs within 5 per cent of the average published prices for A.P.I.-A.S.M.E. code pressure vessels. Where noncode 125-psig. vertical separators are used, the prices shown should be reduced about 6 per cent.

ELEMENTS OF FIELD PROCESSING . . . Part 2

How Pressure Affects Oil and Gas Separation

The author says . . .

1. When selling gas from high-pressure gas-condensate systems, it is seldom possible to justify operating the separator below sales line pressure. This is especially true when the sales line is below 1,000 psi.

2. When compressing the gas is necessary to sell it, compression above the highest pressure of the sales line can seldom be justified. Both initial and operating cost of extra compression will usually be more than the additional liquid revenue.

3. In cycling operations with high-pressure reservoirs it is seldom good to separate at pressures less than 1,200 psig. because of the extra compression costs.

4. On low-gas-oil ratio crude-oil wells, amount of the liquid and its composition are the most important factors.

5. Water dew point, heating value, and other such gas specifications can seldom be met by controlling pressure alone.

6. At high separator pressures, the percentage of methane and ethane in the liquid goes up. To prevent undue losses, handle the separator liquid between that separator and the stock tank with care. Multistage separation is needed, and very often, so is some form of liquid stabilization.

by John M. Campbell

THESE are things that fix the relative amounts of vapor and liquid leaving a separator:

1. Separator pressure.
2. Separator temperature.
3. Well-stream composition.

First two are the only ones that may be controlled directly in field operation, but all must be considered in finding the best conditions. In getting these conditions economics is, of course, the first consideration. Many times the problem is simply to get the greatest amount of stock-tank product to sell. Other times, the gas properties are controlling. Although the conditions vary with the field, all these things must be considered in an over-all evaluation:

1. Amount and value of liquid produced.
2. Amount and value of gas produced.
3. Sales specifications on residue gas including:
 (a) Pressure.
 (b) Temperature.
 (c) Amount of liquefiable components present.
 (d) Heating value.
4. Effect of separator pressure and temperature on the quantity and cost of any auxiliary equipment needed.

On many leases the gas may not be important because it is not sold or the sales specifications may not be too exacting. However, this is becoming less frequent as more gas is marketed and transmission lines increased their specifications.

Item 4 is of particular importance where large amounts of gas are available at moderate to high pressures. This auxiliary equipment includes compressors, dehydrators, liquid-extraction equipment, and the like. Inasmuch as its cost is high compared with separation equipment, the auxiliary equipment is often of major importance. Therefore, over-all economics rather than liquid recovery alone is often the critical factor.

On pumping wells Item 1 is normally the only significant factor.

How to get the relation . . . The relation between pressure, temperature, and composition may be obtained by writing a material balance around a separator. Doing so results in the equation:

$$L_n = \frac{F_n}{K_n (V/L) + 1} \qquad (1)$$

or

$$V_n = \frac{K_n F_n}{(L/V) + K_n} \qquad (2)$$

where:

F_n = mole fraction of any component (methane, ethane, etc.) in the well stream to the separator.
L_n = mole fraction of any component in the separator liquid.
V_n = mole fraction of any component in the separator residue gas.
L = total moles separator liquid.
V = total moles residue gas.
F = total moles of entering well stream.
K = equilibrium vaporization ratio $= V_n/L_n$.

The use of this and other related quantities will be discussed in detail later in the series.

The equilibrium vaporization ratio K is actually a function of all three variables discussed above, while F_n is a composition term proper. These two terms are the independent variables and the final values of L_n and L/V are dependent on them.

The effect of composition on "K" less at pressures below 1,200 psia., the region in which most field separation takes place. For simplicity then, it will be assumed that "k" is dependent only on temperature and pressure. Furthermore, for convenience, each of the

TABLE 1—SUMMARY OF PROCESS CALCULATIONS

	Pressure, psia.			
	400	900	1,200	1,500
Specific gravity gas	0.749	0.716	0.730	0.721
Specific gravity liquid, °A.P.I.	89	141	151	155
Gas-oil ratio, s.c.f. per bbl.	18,100	11,500	10,700	10,100
Total barrels per M.M.c.f.	51.7	78.0	82.8	86.5
Ratio residue gas to well stream	0.939	0.897	0.886	0.878

WELL-STREAM ANALYSIS

Component—	Mole per cent	Component—	Mole per cent
Methane	71.18	Isopentane	0.96
Ethane	15.03	n-Pentane	1.14
Propane	4.78	Hexane	0.92
Isobutane	1.04	Heptanes plus	1.52
n-Butane	3.43	Total	100.00

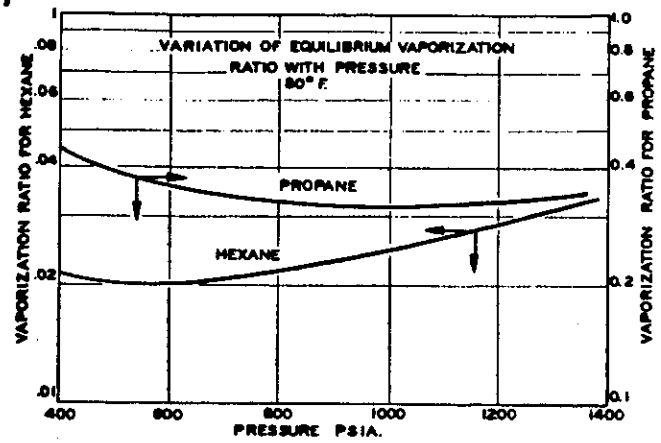


Fig. 1—Regardless of the chart used, a plot of K vs. pressure for a given system at a given temperature gives curves of this general shape.

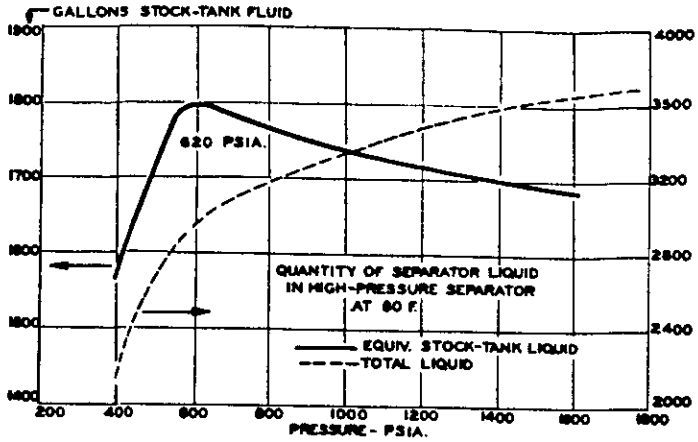


Fig. 3—Effect of pressure on the quantity of separator fluid at 80° F. The solid line is liquid that can be kept in an atmospheric tank at 100° F.

three variables will be discussed separately.

Effect of Pressure on Equilibrium Ratio — K

In most analyses the following components show: methane, ethane, propane, isobutane, n-butane, isopentane, n-pentane, Hexane, and heptanes plus. The latter includes all portions of the liquid heavier than heptane and varies depending on the source of the stream. Values of K for each component have been widely published but it takes experience to find which to apply for a given system. A difference of opinion naturally exists about the use of the various charts available.

Regardless of the chart used, a plot of K versus pressure for a given system, at a given temperature, gives curves of the general shape shown in Fig. 1. Each of the curves reaches a minimum value for K, although the minimum point comes at a different pressure for each component. The minimum points occur in the vicinity of these pressures, when separating at 80° F.

	Psia.
Methane	Above 3,000
Ethane	2,000
Propane	1,000
Isobutane	800
n-Butane	800
Isopentane	700
n-Pentane	700
Hexane	550

Most curves are rather flat near the minimum so that the effective minimum spreads over a range of about 200 psi.

From Equation 1, you can see that the amount of liquid obtained will go up as the value of K goes down, if all other factors remained the same. By using Fig. 1, when separating a mixture of ethane and hexane at 80° F. the best pressure should be between 600 and 1,000 psia. The actual value would depend on the relative amount of each.

An actual multicomponent hydrocarbon mixture, as produced, is very complex; and the pressure at which the most liquid is formed is not easily found. Not only does the composition vary but the lowest value of K differs from the several components.

Retrograde Condensation

Fig. 2 is a typical phase curve for a multicomponent hydrocarbon mixture when plotting temperature against pressure, the shape of the curve depending on composition of the system. At all points inside the "envelope" both liquid and gas are present, so this is the region under study when considering separation. The dashed lines show the pressure and temperature at which the ratio of liquid to gas is constant.

In most of the two-phase region, lowering the pressure at constant temperature results in less liquid. However, such a plot for most hydrocarbon mixtures has a region where decreasing pressure results in more liquid. This is shown as the isothermal retrograde region—a region where the reduction

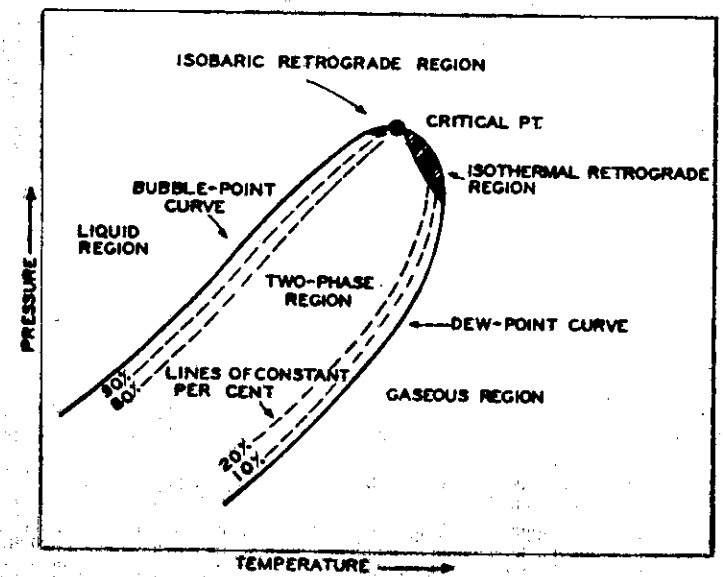


Fig. 2—A typical phase diagram for a multicomponent mixture on pressure-temperature coordinates. Both liquid and gas are present under the envelope.

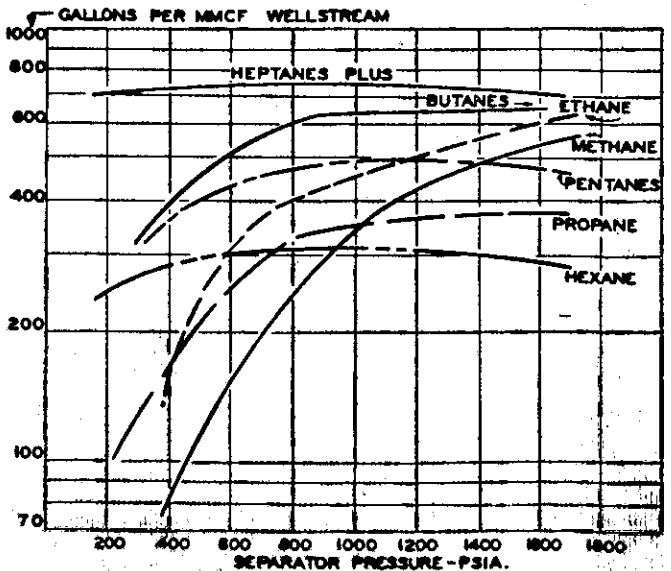


Fig. 4—Variation of separator fluid with pressure at 80° F. The heptanes plus stay about the same while the other components vary widely.

of pressure at constant temperature results in more liquid. Thus the term retrograde, or "contrary to general fact" is applied when this occurs.

Experience shows that retrograde effects are present in the production and separation of oil and gas. This is illustrated in Fig. 1 by the shape of the curves. As the pressure is lowered on a mixture consisting largely of hexane, the amount of liquid goes up until pressure reaches about 550 psia. at which point it begins to go down.

Effect of Pressure on Amount of Liquid

Fig. 3 gives the amount of separator fluid obtained when varying the separator pressure at constant temperature (80° F.). The solid curve shows only that quantity of the total liquid that could be held in an atmospheric tank at 100° F. The dashed curve shows total liquid.

At a different temperature the curves would be displaced upward or downward. With a different well-stream composition the shape of the curves and the point of maximum stock-tank liquid would vary some. But the curves shown are typical.

Below the maximum point the slope of both curves becomes large, for large amounts of the "heavy ends" pass out with the residue gas and little of the methane and ethane are condensed. Above the optimum pressure the total amount of liquid continues to rise for the increased condensation of methane, and the lighter ends more than make up for the lower amount of the hexanes plus. The greater loss of heavy ends is shown by the solid line which gradually goes down.

Although not shown, the total quantity of liquids would reach a maximum about 1,900 psia. (see Fig. 1, Part 4, to follow). At this point the amount of stock-tank liquid would go down more rapidly.

These variations are shown in Fig. 4. Marked changes are noted for all components except the heptanes plus, which stay about the same. Below 600 psi., the amount of all of the components goes down. That accounts for the rapid decrease shown in Fig. 3. At higher pressures methane and ethane go up rapidly, the amount of propane and butane remains about the same, and the others go down slightly. Since about 85 per cent of the stock-tank liquid is made up of the pentanes plus, the decrease in stock-tank liquid at high pressures may be expected.

ELEMENTS OF FIELD PROCESSING . . . Part 3 of a Series

How Temperature Affects Separation

by John M. Campbell

These rules of thumb on temperature in separation generally hold true:

1. When separating only to get stock-tank fluid for sale as crude oil and when residue gas is salable the amount of liquid is usually at the optimum between 15° and 30° F.

2. On small leases producing less than 2 M.M.c.f.d. the return per dollar of investment is often greater at separation temperatures of 40°-60° F.

3. The recovery of propane and butane by refrigeration alone is seldom justified economically if the temperature needed is less than —15° F.

4. Separation at a temperature above 85° F. is seldom justified.

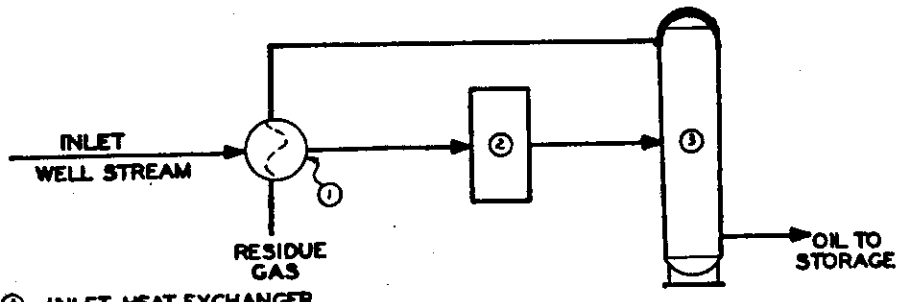
VERY often temperature is the only effective variable we can control in the separation of oil and gas. The composition of the gas is largely out of the control of the operator, while the pressure is usually set by the sales pressure.

Control of temperature almost always involves cooling, for well-stream flowing temperatures are normally above the best separation temperature. We can get cooling by any one or more of these:

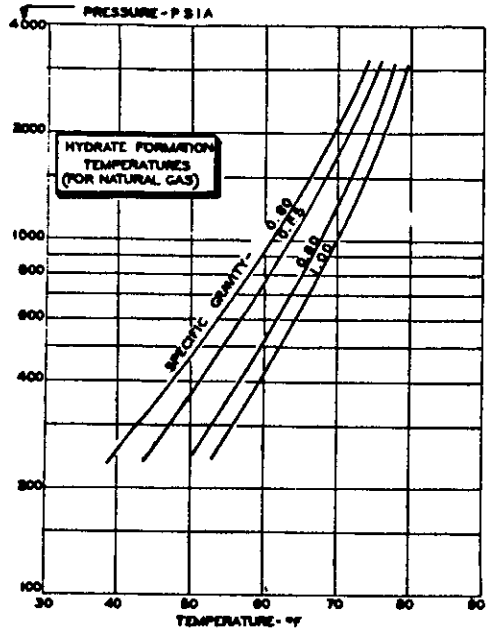
1. Atmospheric or forced-draft air cooling.
2. Water-gas heat exchanger using a cooling tower or evaporative-type cooler.
3. Refrigeration.
4. Expansion of the well stream across a choke or valve under substantially adiabatic conditions.

Of course, the first is limited by atmospheric temperature, so it cannot be used widely in the Southwest or Gulf Coast area. Some forced-draft air coolers bring the well stream to within 10° F. of the dry-bulb temperature, but "standard" units are for a 20° F. approach. Atmospheric units normally give a 25°-35° F. approach to the atmospheric dry-bulb temperature.

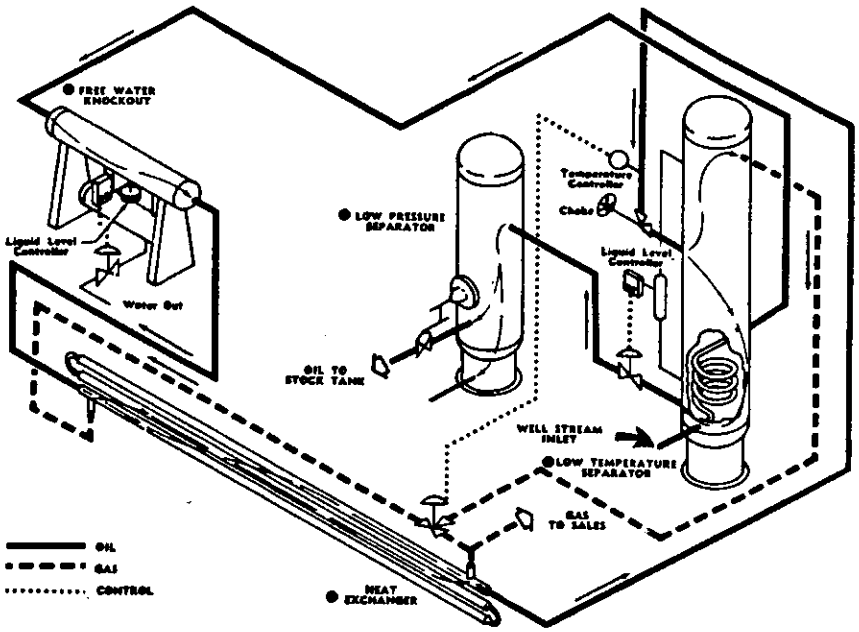
Some typical temperatures . . . In heat-exchanger design, costs require that the outlet water temperature be within 10° F. of the inlet well stream. The inlet-water temperature depends not only on the method of cooling but also on atmospheric conditions. In general, though, the inlet water to the heat exchanger can be cooled within 7° F. of the atmospheric wet-bulb temperature.



SIMPLE LEASE HOOKUP for lowering well-stream temperature above hydrate point. Fig. 2.



TYPICAL hydrate-formation temperatures for natural gas. Fig. 1.



FLOW DIAGRAM for typical low-temperature separation system. (Courtesy Black, Sivalls & Bryson, Inc.) Fig. 3.

On this basis a water heat exchanger and an induced-draft cooling tower will give these typical well-stream temperatures:

Area—	Temp. (°F.)
Lafayette, La.	100
Houston	96
San Antonio	93
Abilene, Tex.	91
Kilgore, Tex.	95
Oklahoma City	92
Amarillo, Tex.	87
Rocky Mountain area	81

All of the above should not be exceeded over 5 per cent of the time during the summer months. The winter months, of course, usually are no problem.

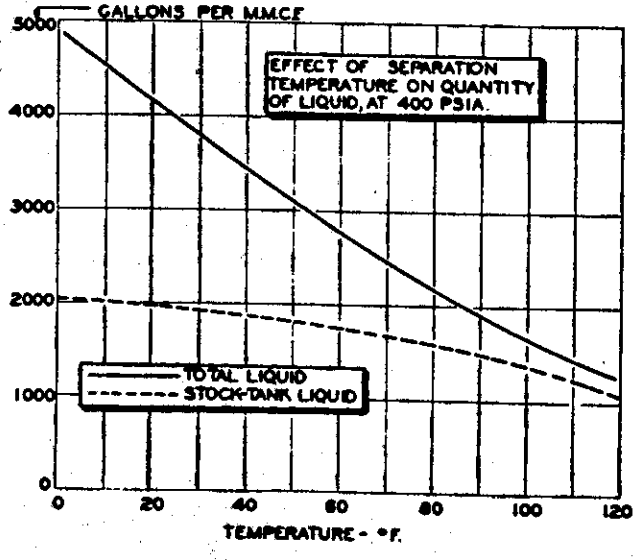
Refrigeration types . . . Refrigeration is a subject within itself and involves a comparison of the types available and their limitations. The temperature we can attain with refrigeration have no effective limits. But there are many practical ones. For instance, if the separator temperature is below the hydrate formation temperature, special measures are necessary to combat the hydrates.

Below — 15° to — 20° F. steel rapidly loses its ductility due to carbon embrittlement. Lower temperatures make necessary low-carbon steel or special metals. Consequently, few units are designed to operate at lower temperatures unless they are for ethane recovery.

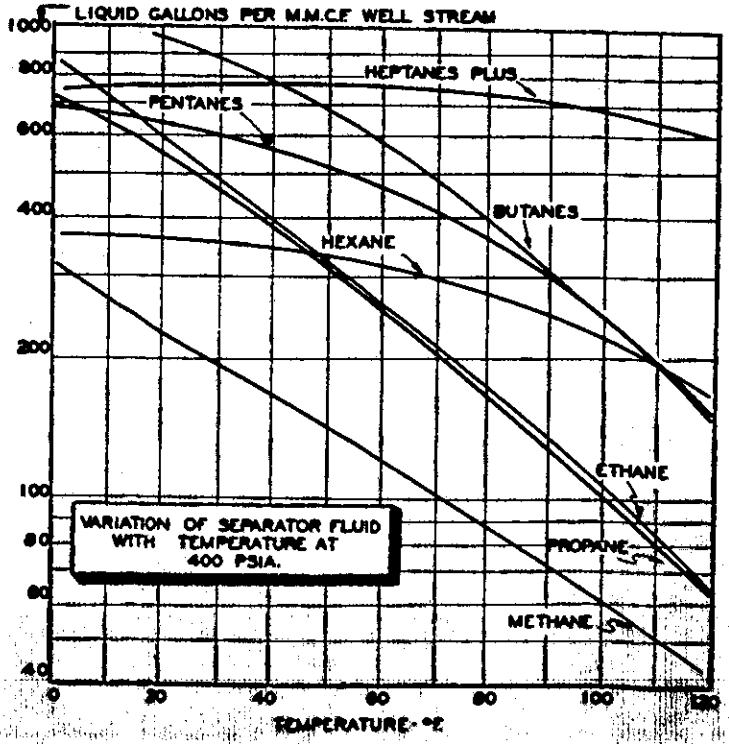
Expansion is the most common of the cooling systems because the high-pressure gas is becoming more common. Also little capital outlay is involved with an expansion system.

Hydrate Formation

The hydrate formation temperature mentioned above depends on pressure. It is the temperature at which a portion



QUANTITY of separated liquid as a function of temperature, at 400 psia. Fig. 4.



VARIATION of separator fluid with temperature at 400 psia. Fig. 5.

of the gas is chemically combined with water present in the gas to form hydrates. These hydrates differ in structure from ice; but they are solid under hydrate conditions and may block a line or separator. They are opaque white when solid. When melted they form a milky hydrocarbon-water emulsion. In fact, the presence of hydrates may often be determined by removing pressure and draining the vessel.

Hydrate formation is not fully understood. However, a correlation such as Fig. 1 is useful. Each curve is actually a band rather than a line and applies only to gases of "normal" combustion. Sour gases and those of abnormal composition might show different action; but the curves shown are good for most calculations.

System Characteristics

Figs. 2 and 3 are flow diagrams of typical systems for reducing temperature. They are very similar except that the low-temperature system is, of course, more complex. The dividing line between the two is essentially the hydrate-formation temperature, the former having application only when no hydrates are formed.

In Fig. 2 the separator is the conventional oil-field type. The heat exchanger is normally of the pipe-in-pipe style but shell-and-tube types are applicable. Its purpose is to cool the inlet well stream with the cold residue gas from the separator in order to reduce the amount of cooling at Point 2.

The flow diagram in Fig. 3 shows one form of the equipment devised for operating a separator below the hydrate point.

This particular system handles the hydrates formed by melting them, using a coil in the bottom of the separator. The heat necessary may be furnished by the warm inlet well stream (as shown) or by a water heater or steam generator.

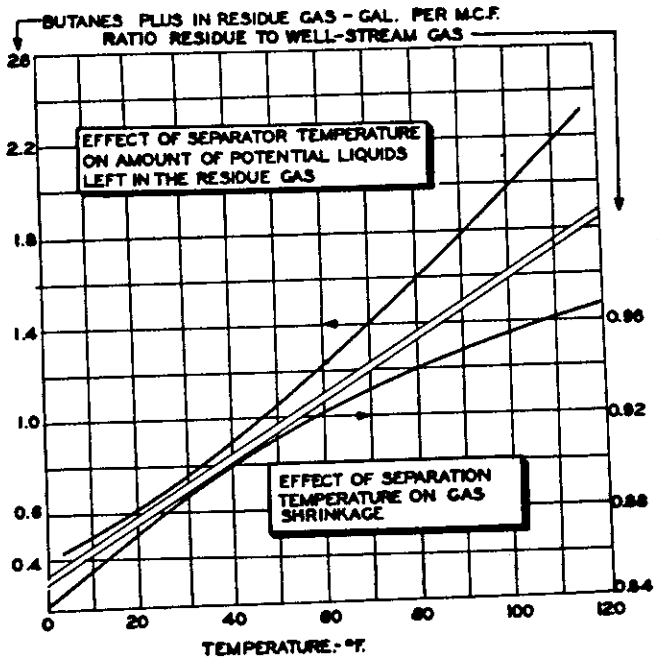
Where little free water is produced, the free-water knockout is sometimes eliminated. When this is done the systems shown in Figs. 2 and 3 become very similar except for the separators.

Another form of low-temperature-separation system prevents hydrate formation by the addition of an inhibitor. This is usually a dilute solution of one of the glycols, although alcohol is sometimes used.

Effect of Temperature on Liquid

To illustrate effect of temperature on the liquid and gas, refer to the well-stream analysis at 400 psia. in Table 1, Part 2. Temperatures ranging from 120° to 0° F. will be applied to this mixture for our study here.

There has been a tendency on the part of many to operate the separator as close to 0° F. as possible without regard to the economics involved.



EFFECT of separator temperature on the residue gas. Fig. 6.

In many cases where only stock-tank fluid is desired, the optimum temperature is actually between 20° and 25° F. A complete analysis on the system used here, for example, gave an optimum temperature of 21° F. at 900 psia.

This is partially explained by Fig. 4. It shows the variation in the quantity of stock-tank fluid and total fluid with temperature at 400 psia. In lowering the temperature from 120° to 0° F. the amount of stock-tank liquid about doubles while the total liquid is increasing about four times. Any liquid condensing that is not marketable reduces the quantity of salable residue gas accordingly. The approximately 3,000 gal. of nonstock-tank liquid shown at 0° F. in Fig. 4 has a market value, as gas, of about $15 based on a sales price of $0.12 per M.c.f.

Fig. 4 shows that there will be little additional stock-tank revenue by reducing temperatures much lower than 0° F. In fact, from 20° F. down the curve becomes awfully flat. All calculations assume that the only product desired is one that may be held in atmospheric stock tanks. In the event propane or L.P.G. or both are desired as products, the curve would not be so flat.

Fig. 4 is verified by the results shown in Fig. 5, a breakdown of the separator liquid by components. The curves clearly show effect of temperature on quantity of the various hydrocarbons. The lighter hydrocarbons (methane, ethane, and propane) increase rapidly with decreasing temperature. This accounts for the rapid increase in total separator fluid.

Temperature has little effect on the quantity of the heptanes plus. Consequently, where the final fluid is composed primarily of "heavy" ends, there is little advantage in decreasing temperature.

Note, though, that there is a rapid increase in the heptanes plus and hexane. It is these which largely account for the increase in stock-tank fluid shown in Fig. 4. Consequently the ratio of pentane and hexane to the heptane plus fraction has a great effect on the percentage increase of the stock-tank liquid with decreased temperature. No specific correlations are available but this has been shown to be generally true.

In an increasing number of cases the desired gas properties determine the choice of the separator temperature. Therefore, it is often necessary to give consideration to water dew point, hydrocarbon dew point, and heating-value specifications.

In most instances optimum separation temperature is lower than needed for water dew-point requirements. Therefore, there is no problem in that regard. However, temperatures less than +10° F. are often required to satisfy hydrocarbon dew-point and heating-value specifications. This is illustrated by the upper curve of Fig. 6, which is typical of that for a "rich" well stream. As might be expected, the heating-value curve has the same general slope.

Heating-value specifications are a different problem with gases containing large quantities of inerts such as nitrogen, carbon dioxide, and helium. The quantity of liquid from such gases is often relatively small, and gas revenue is more important. Consequently, low temperatures might not be feasible.

The extra liquid condensed might lower the heating value below the minimum specified, or lower the sales value by the imposition of a heating-value penalty clause.

Loss of revenue due to gas shrinkage is, of course, a problem in many cases. Lower curve of Fig. 6 illustrates results.

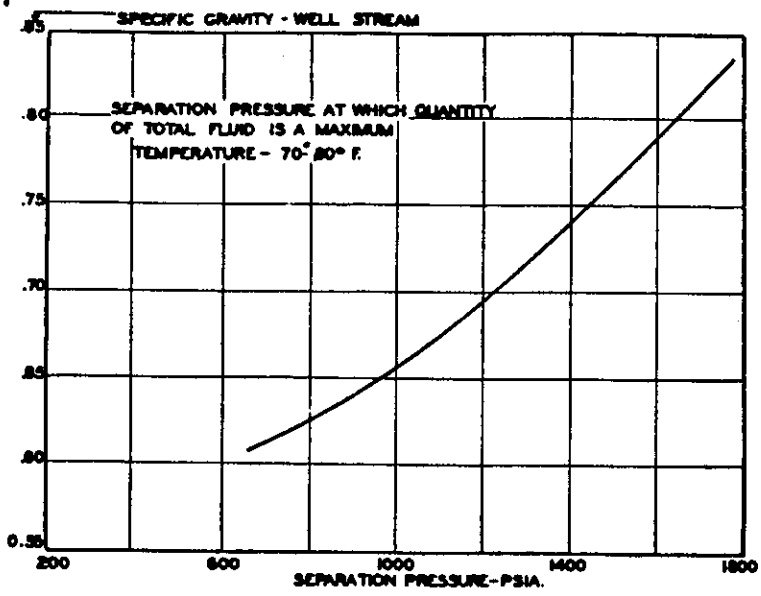


Fig. 1.

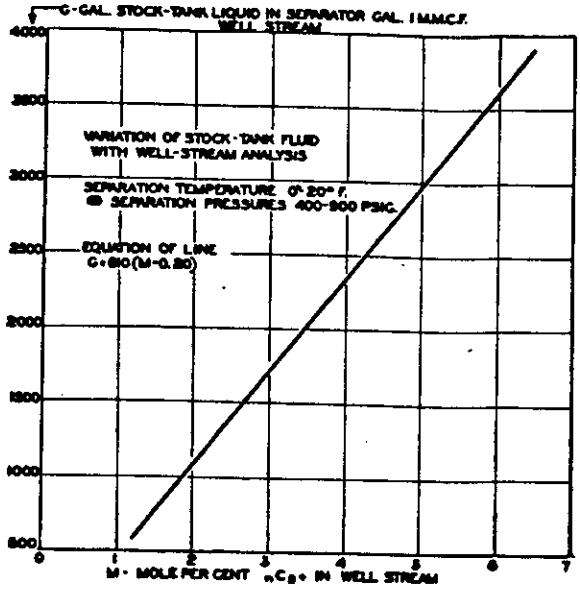


Fig. 3.

ELEMENTS OF FIELD PROCESSING

How Well-Stream Composition Affects Degree of Separation

Part 4 of a Series

by John M. Campbell

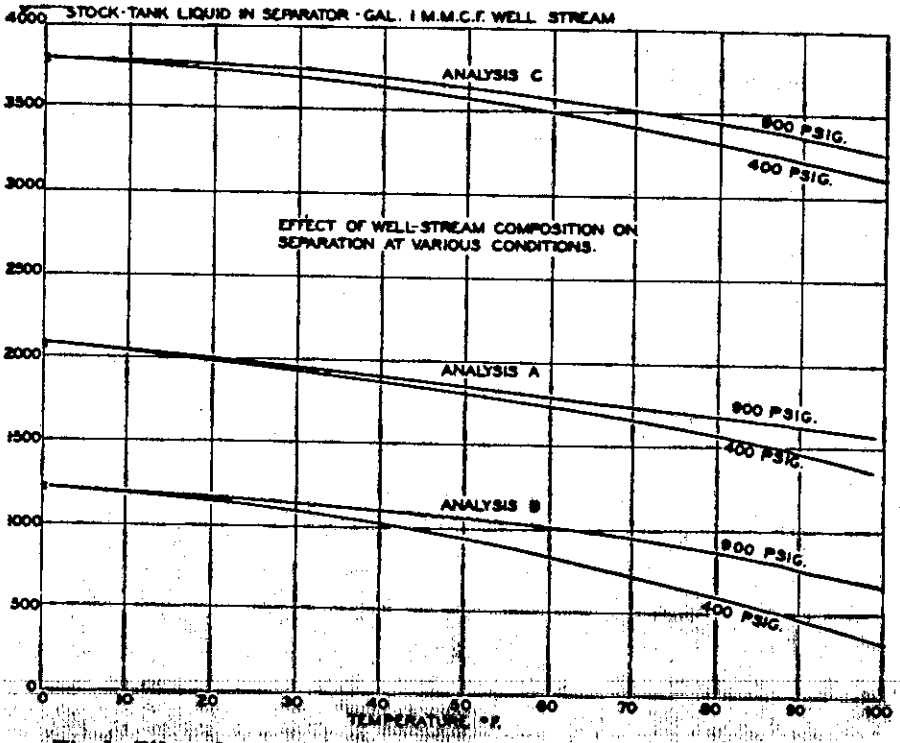


Fig. 2—Effect of composition and temperature on quantity of stock-tank fluid.

The Author Says

- The mole percentage of n-pentane plus in the well stream is the prime composition factor determining the quantity of stock-tank fluid in the separator.
- The pressure at which the maximum amount of total separator fluid is obtained, may be correlated against the specific gravity of the wellstream.
- Below 20° F. composition is the prime factor affecting the quantity of stock-tank fluid in the separator.
- Any increase in recovery due to lowering temperature depends on the well-stream composition.

WELL-STREAM composition is the third and equally important variable affecting the degree of separation between oil and gas. In any given separation problem its effect may never be completely separated from temperature and pressure effects (discussed in Parts 2 and 3), but it is enlightening to consider the effect of composition when all other factors are constant.

Composition is, of course, the basic variable that determines how much potential liquid is available and the economics of producing a given product. A complete analysis of the effect of composition requires that the

amounts of the various components be known. This entails sampling and analysis by low-temperature fractional, spectrographic, or equivalent method.

For normal separation it is desirable to know the percentage of methane, ethane, propane, isobutane, normal butane, isopentane, normal pentane, hexane, and the heptanes plus fraction as well as the molecular weight and specific gravity of these latter "heavier ends." Often it is more convenient to sample and analyze the liquid and gas from a separator rather than the incoming well stream. In this case it is also necessary to determine the gas-oil ratio, pressure, and temperature in the separator so that a composite well-stream analysis may be calculated.

Other data . . . Obtaining the above component analysis is relatively expensive, particularly if a large number of wells are involved. Consequently, data from other sources may often be employed for preliminary calculations. Data of this type include that from N.G.A.A. and C.N.G.A. charcoal and compression tests and specific gravity determinations. Data of these types must be used with caution for they often prove misleading.

The results of such tests are usually expressed in gal. per M.c.f. (gallons of total liquid per 1,000 standard cubic feet). The N.G.A.A. compression and charcoal tests supposedly purport the amount of stock-tank gasoline present in the gas stream. Either will serve as a preliminary test and to a full-scale program.

As in previous installments, the scope here is limited to the production of stock-tank fluid. Such fluid is normally composed primarily of the pentanes plus. Therefore it is natural that the quantity of such components present in the well stream would have a strong influence on the quantity of stock-tank fluid.

The richness of a well stream is often expressed in gal. per M.c.f. of propane plus which does not necessarily reflect the amount of recoverable liquid. Casing-head gas, for example, often contains large quantities of propane yet the recoverable liquid is quite small.

The total quantity of separator fluid may often be expressed as a function of the well-stream specific gravity. Fig. 1 shows the pressure at which the maximum quantity of separator liquid is obtained for a given specific gravity well stream. The curve shown indicates the average value obtained from a number of well-stream analyses with an average deviation of ±5 per cent. The curve shown has application where maximum removal of liquids is important and serves to show the combined effects of pressure and composition.

TABLE 1

Component—	Mole per cent — Analysis A	Analysis B	Analysis C
Methane	71.18	70.01	73.54
Ethane	15.03	15.03	15.03
Profpane	4.78	7.42	1.20
Isobutane	1.04	1.04	0.87
N-butane	3.43	3.43	2.73
Isopentane	0.96	0.96	0.36
N-pentane	1.14	1.14	0.45
Hexane	0.92	0.63	1.04
Heptanes plus	1.52	0.34	4.78
	100.00	100.00	100.00
Gal./M.c.f. C_3+	4.64	4.64	4.64
Mole. % n-C_5+	3.58	2.11	6.27
Mole. wt. C_7+	115	115	115
Sp. gr. C_7+	0.71	0.71	0.71

Effect of composition . . . In order to illustrate the effect of composition, three analyses have been chosen. These are shown in Table 1. In each case the gal. per M.c.f. of the propane plus fraction is the same although the pentane-plus fraction varies markedly. In each case the pentane plus fraction is assumed to have the properties of octane.

Fig. 2 summarizes the quantity of stock-tank fluid obtained in the separator for various separation conditions. The curves obtained are typical of those for any analysis. Several conclusions may be drawn from examination of these curves.

Although the gal. per M.c.f. of the propanes plus is the same, the recovery varies markedly and would appear to be some function of the quantity of pentanes plus.

Below 40° F., and particularly below 20° F., the curves are quite flat. Also, below 20° F. the recovery for a given gas is practically the same regardless of whether the pressure is 400 or 900 psig. Above 20° F. pressure begins exerting an effect on recovery.

The results shown in Fig. 2 are typical of those obtained with a large number of analyses. These may then be used to predict the effect of temperature, pressure, and composition on separation.

The effect of composition on the quantity of separator stock-tank fluid below 20° F. may be expressed by Fig. 3, where the prime variable is the mole per cent normal pentane plus in the well stream. The equation of this line is

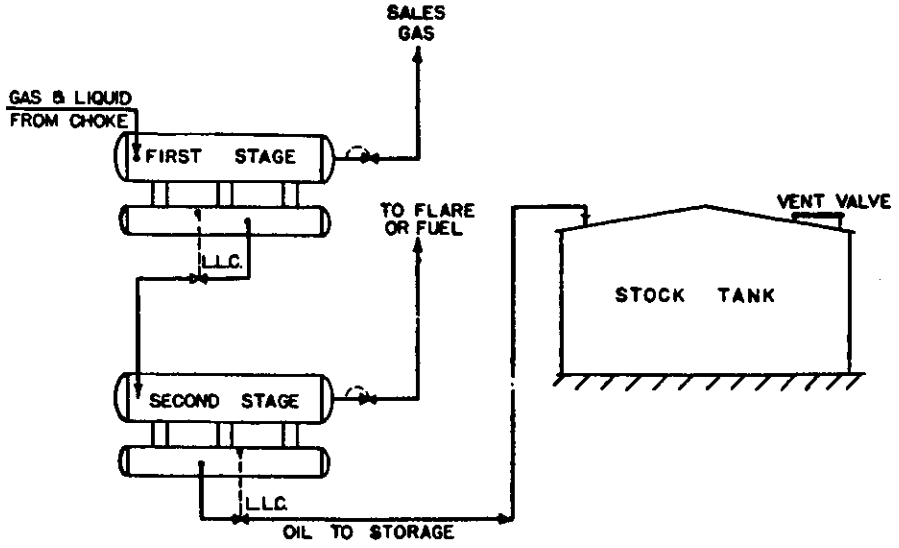
$$G = 610\ (M - 0.20) \qquad (1)$$

where

G = gal./M.M.c.f. stock-tank fluid in the separator

M = mole per cent n-pentane plus in the well stream

From this equation the quantity of separator stock-tank fluid may be easily predicted from the well-stream analysis as long as the separation temperature is below 20° F. Above 20° F. a more complex equation would be needed. Work is now under way in developing a more general correlation.



STAGE SEPARATION of oil and gas from the well. Fig. 1.

ELEMENTS OF FIELD PROCESSING

Handling of Separator Liquid—Stage Separation

Part 5 of a Series

by John M. Campbell

THE previous discussions have considered those factors affecting the quantity of separator liquid. It is well then to consider the methods of placing this liquid in the stock tank.

Regardless of the method used, not all of the potential stock-tank fluid in the first-stage separator will be retained in the stock tank. The most efficient method, stabilization, will be discussed in Part 6.

Stage separation . . . This is the name given to the alternative system shown in Fig. 1. The liquid from the first stage is expanded into a separator at lower pressure, in one or more steps, until reaching the stock tank. Each separator is considered one stage, as is the stock tank. The flow sheet in Fig. 1, therefore, represents three-stage separation.

Stage separation is used for two basic purposes:

1. To increase stock-tank recovery by minimizing vaporization.

2. To remove the bulk of the vapors formed when the pressure is lowered so that they will not enter the stock tank.

The latter is often quite critical for the evolution of large quantities of vapors in the stock tank causes the liquid to "roll," thus increasing "weathering" losses. Many tests have shown that such turbulence is one of several factors that increase tankage losses.

It is obvious that the chief problem with stage separation is the determination of the number of stages and the pressure at which to operate each. The number of stages needed depends on the first-stage pressure and liquid composition as primary variables. Under normal conditions, though, it is usually difficult to justify over four stages because of the economics involved. As more stages are added the amount of stock-tank liquid increases, but usually not enough to pay out the cost of the additional separator(s) in a reasonable period of time.

How many stages? . . . The question still exists as to whether to use two, three, or four-stage separation. In the absence of actual tests this may only be accurately determined by a set of tedious calculations. In the absence of either, the rule of thumb stating that the pressure ratio per stage should not exceed:

5—for gas-condensate production
7—for crude-oil separation where the stock-tank oil gravity is greater than 50° A.P.I.
9—for crude-oil separation where the stock-tank oil gravity is less than 40° A.P.I.

is often useful. Like all rules of thumb, exceptions will be found.

After selection of the number of stages, it is necessary to determine at what pressures the intermediate stages should be operated. Based largely on crude-oil data, the following formula has been proposed:

$$R = (P_1/P_s)^{1/n} \qquad (1)$$

$$\text{or } P_2 = P_1/R = P_s R^{n-1}$$

$$\text{and } P_m = P_s R^{n-(m-1)} \qquad (2)$$

where:

n = number of interstages
R = pressure ratio
P_1 = first-stage pressure, psia.
P_2 = second-stage pressure, psia.
P_m = pressure, psia., for the last intermediate stage used
M = number of the stage
P_s = stock-tank pressure, psia.

Application of Equations 1 and 2 to a three-stage separation problem were $P_1 = 400$ psia. and $P_s = 14.2$ psia. would give

$$R = (400/14.2)^{1/2} = 5.30$$

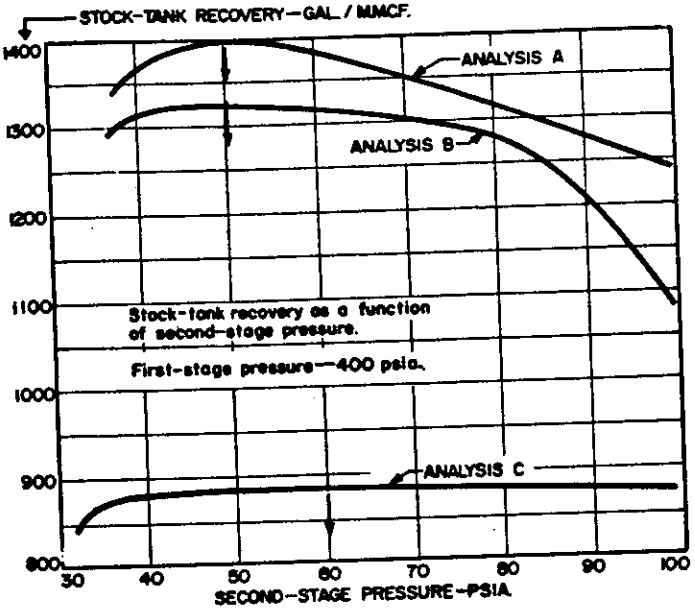
$$P_2 = (14.2)\,(5.30)^{2-1}$$

$$= 75.3 \text{ psia.} = 61.1 \text{ psig.}$$

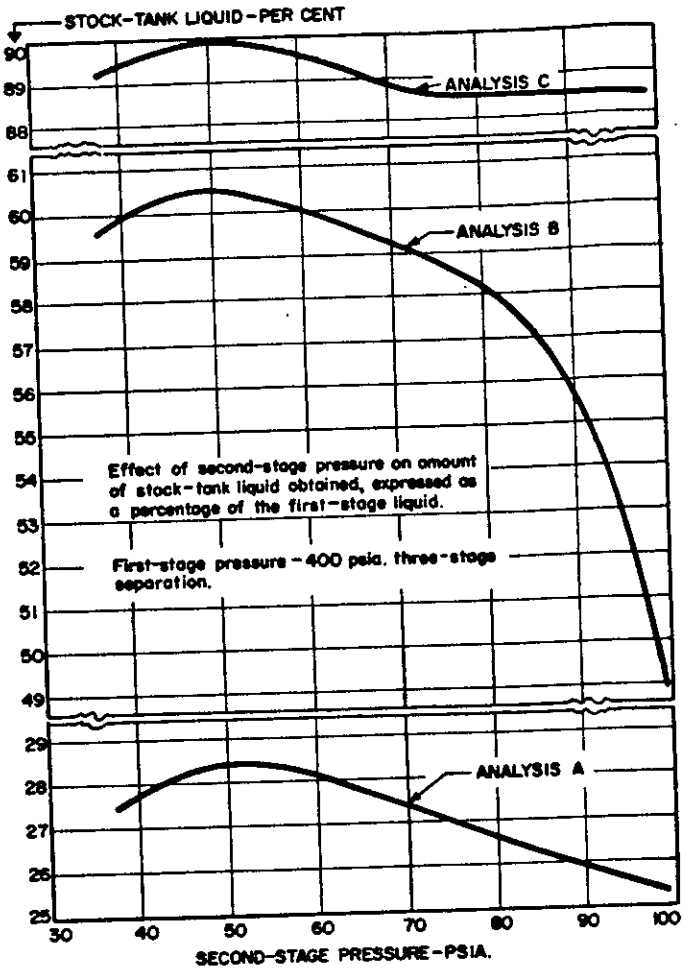
Equation accuracy . . . As might be expected there are many instances where the use of rigorous calculation methods will not check the results given by Equations 1 and 2. These equations, of course, assume that the ratio per stage should be constant. Examination of vapor-liquid equilibrium constants, though, does nothing to support this assumption. In fact, complete analysis of a separation problem often shows that the ratio between the last stage and the stock tank is considerably smaller than that between the other stages.

There is also no provision in this formula for differences in first-stage liquid analyses, which obviously affect the method of handling. In the absence of a complete analysis, however, Equations 1 and 2 provide a quick and easy means of estimating interstage pressures. From a practical standpoint it is often relatively easy then to determine optimum pressures by field experimentation.

Effect of increasing number of stages.



STOCK-TANK RECOVERIES from stage separation of three liquids from Table 2. Fig. 2.



FRACTION OF FIRST-STAGE liquid recovered in the stock tank. Fig. 3.

Table 1 shows some of the data obtained by Whitely[1] illustrating the effect of increasing the number of stages on 13 different well streams. In going from two to four-stage separation recoveries were increased from about 3 to 22 per cent, depending on well-stream composition, with an average increase of 8 per cent. This wide variation illustrates the futility of relying on rules of thumb alone in determining the optimum number of stages.

Examples

The factors that effect stage separation may best be illustrated with a concrete example. Consequently the analyses shown in Table 2 were chosen for study. These represent the analysis of a liquid present in the first stage of separation. Analyses A and B represent liquid separated from the same well stream at 385 psig., the only difference being in the temperature of separation. This has been done to illustrate the effect of the original temperature of separation upon the efficiency of subsequent stage separation.

Analysis C differs from the other two in that it contains relatively little of the components ethane through pentane, by comparison with the other two analyses.

TABLE 1—INCREASE IN STOCK-TANK LIQUID RECOVERY BY INCREASING NUMBER OF SEPARATION STAGES
(Separator temperature, 80° F.)

Test No.	Stock-tank liquid rec. 2-stage separation bbl./ M.M.c.f.	Stock-tank liquid rec. 4-stage separation bbl./ M.M.c.f.	Per cent increase in stock-tank liquid rec. (4-stage over 2-stage)
12	66.0	78.0	18.20
15	11.5	12.2	6.09
16	9.5	9.8	3.16
19	14.7	15.4	4.75
21	35.9	37.1	3.34
28	57.5	62.5	8.70
29	26.2	28.5	8.78
30	26.3	32.1	22.00
31	26.7	30.2	13.10
33	15.3	16.2	5.89
35	15.0	15.5	3.43
38	23.3	24.1	3.59
39	17.0	17.8	4.71

Average per cent increase in stock-tank liquid recovery by adding two intermediate stage separators = 8.14 per cent.

TABLE 2—SUMMARY OF SEPARATOR LIQUID ANALYSES

Component—	Mole per cent Analysis A	Analysis B	Analysis C
Methane	11.91	8.26	10.18
Ethane	21.29	10.60	2.72
Propane	16.86	9.86	3.48
Isobutane	4.98	4.03	2.16
n-Butane	17.79	16.34	2.76
Isopentane	5.44	7.13	1.77
n-Pentane	6.61	9.24	2.20
Hexane	5.64	11.30	*74.73
Heptanes plus	9.48	23.24	
Total	100.00	100.00	100.00
Separation pressure, psig.	385	385	385
Separation temp., °F.	0	80	80
Total first stage separator liquid, gal./M.M.c.f.	4,923	2,193	984

*Hexane plus.

Effect of intermediate stage pressure on recovery . . . Fig. 2 summarizes the stock-tank recoveries to be expected when subjecting the three liquid analyses in Table 2 to two additional stages of separation.

As might be expected, Analysis A gives the highest total recovery because more liquid was present in the first stage separator, per million cubic feet of well stream processed. Although the shape of the curves is different they all pass through a maximum, i.e., the intermediate pressure at which maximum liquid recovery is obtained.

It is obvious though that the intermediate pressure is not critical in the case of Analysis C, a pressure anywhere between 40 and 100 psia. offering a practical solution.

The more pronounced variations shown by Analyses A and B are typical of analyses exhibiting relatively large quantities of pentanes. Pentane is normally a stock-tank component, yet is relatively easy to vaporize. The reduction in recovery at lower pres-

sures is largely due to the loss of this pentane. This phenomenon is particularly pronounced when large quantities of methane and ethane are present.

Below 30 psia. the recovery drops off rapidly. Elimination of the intermediate stage in the case of Analysis B, for example, reduces the recovery to 1,105 gal. per M.M.c.f.

As the curves show, the practical maximum occurs throughout a small range instead of a single pressure.

Effect of composition and pressure on efficiency of recovery . . . Fig. 3 shows the fraction of the first-stage liquid recovered in the stock tank. Here it is readily seen that Analysis A, although showing the highest recovery (Fig. 2), has the greatest loss. On the other hand, stage separation of Analysis C is very efficient.

Several conclusions may be drawn from these results, which have been borne out by field experience. First, stage separation of liquids containing large quantities of methane through pentane usually show large losses when processed by stage separation. Typical of this is the fluid produced when processing gas-condensate wells by means of low-temperature separation.

Second, first-stage liquid in which the potential stock-tank fluid is composed largely of the hexanes plus may usually be efficiently processed by stage separation. Most crude-oil wells producing 45° A.P.I. crude, or less, in the stock tank fall into this category.

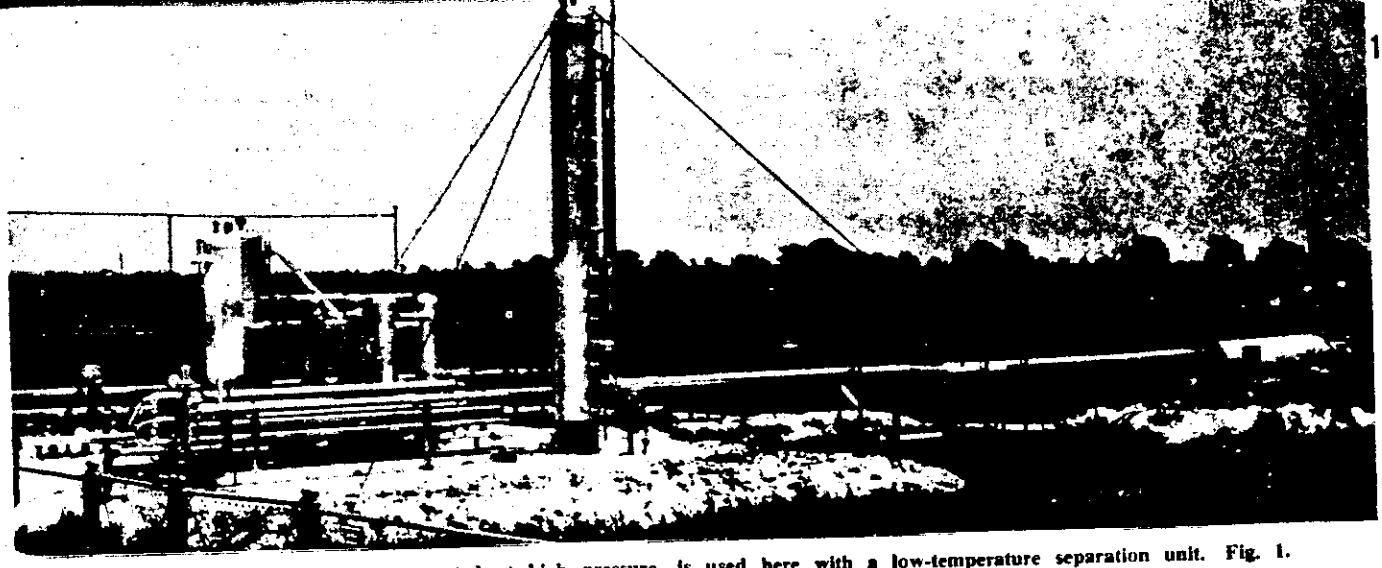
The type of plot represented by Fig. 3 is not only valuable in estimating the efficiency of stage separation but is also convenient to use in determining the optimum pressure since fluctuations in recovery are more pronounced.

Inasmuch as there are no correlations at present that accurately give the number of stages needed, as well as the optimum operating pressures, the above results may only be used as a general guide. Practical considerations require that these answers may only be accurately obtained by flash vaporization calculations, verified by field test. The current widespread use of electronic calculators makes this a reasonable and economical approach.

The economics of using various numbers of stages may be determined by employing the cost estimation curves in Part 1. The total cost is obtained by simply adding up the individual separator costs shown. If a pre-piped skid-mounted unit is desired, about 10 per cent should be added to the above sum.

Reference

1. Whiteley, Burton, M.S. thesis, University of Tulsa, 1947.



COLD-FEED STABILIZER, operated at high pressure, is used here with a low-temperature separation unit. Fig. 1.

ELEMENTS OF FIELD PROCESSING—6

Stabilization of Separator Liquid

by Dr. John M. Campbell

THE processing of separator liquid by stabilization has become increasingly common in the past 5 years primarily because of the widespread use of low-temperature separation equipment. However, more use is being made of stabilization where normal temperatures prevail even though it is harder to justify the higher expense.

A stabilizer is nothing more than a fractionation column designed to selectively separate the components without undue loss of salable components. In many details it is identical to the fractionation towers used in gasoline plants and refineries. In general, though, it utilizes simpler controls and a direct gas-fired source of heat.

Desirable features . . . Any stabilizer for field use should generally have the following features:

- Be self-contained in that the only utilities needed are those readily available, i.e., natural gas and sometimes water and/or electricity.
- Be reliable and capable of operating continuously with only periodic attention.
- Have a simple and reliable control system that may be easily adjusted, and to a certain extent maintained, by the average lease pumper and switcher.
- Be able to handle widely fluctuating quantities of fluid of varying composition.
- Be readily installed and dismantled in the field.

General Characteristics

There are three basic types of stabilizers now being used:

1. Conventional reflux type, operating at pressures from 150 to 300 psia., using water or refrigeration-cooled reflux condensers.
2. High-pressure nonrefluxed "cold-feed" stabilizers, operating at pressures of 150-300 psia., with the feed entering at or near the top.
3. Same as (2), except that the operating pressure is from 40 to 70 psig.

There is nothing unique about the first type. It is nothing more than a regular gasoline-plant-type tower adapted for field use. The reflux system complicates the control system somewhat, increases cost, and usually requires the availability of water. Consequently, types (2) and (3) are used wherever possible.

Cold-feed units . . . The cold-feed stabilizers have many limitations, but they are ideally suited for many field stabilization problems. Consequently, a large number of them are now in use. The remainder of this article will concern this type since refluxed stabilizers have been widely discussed in the literature.

Fig. 1 is a view of a high-pressure cold-feed stabilizer used in conjunction with a low-temperature separation unit in the Gulf Coast area. The liquid from the low-temperature separator enters the tower about 2 ft. from the top, the overhead line shown carrying the gas evolved in the tower. Heat is furnished by a gas-fired heater similar to a standard oil-field heater, except that Du Pont "Hi-Tec" salt is used as the liquid instead of water.

Fig. 2 is a schematic sketch of such a stabilizer with an inset showing a common method of handling reflux when such is needed.

The head necessary to force the liquid through the salt-bath heater is furnished by taking said liquid off a chimney tray part way up the tower. This has the obvious disadvantage of increasing tower height.

Fig. 3 is a schematic sketch of a low-pressure cold-feed stabilizer using a 15-psig. gas-fired steam generator as a source of reboiler heat. In this unit, the reboiler may either be incorporated in the base of the tower or be a separate vessel.

Stabilizer Design

The cold-feed stabilizer in reality is nothing more than a stripping column in which the feed is cold enough to prevent undue loss of stock-tank components overhead. Obviously a flash takes place in the top of the tower that to some degree is comparable to a flash in a separator. So there is obviously a limit as to how high the feed temperature may be without encountering undue losses overhead.

Since the losses gradually increase as feed temperatures are raised, there is no one critical temperature. But field experience has shown that the temperatures should be not greater than those shown in Fig. 4, for best results.

From Fig. 4 it may be deduced that the high-pressure tower experiences a lower stock-tank-liquid loss than the low-pressure unit, with the same feed temperature. Field experience has borne this out, and it is possible to reach the same conclusion by examination of vapor-liquid equilibrium ratios.

Tower operation . . . It is really necessary to understand what takes place in the tower to realize the limitations and general characteristics of the unit. The cold feed is fed on or near the top tray of the tower where it contacts the vapor rising from the tray below. Because the liquid is much colder than this vapor, a very unstable condition exists. Heat exchange takes place between the two streams, and part of the liquid, including heavy ends, vaporizes while a portion of the vapor is condensed.

Because of the large temperature differential, causing this instability, a good separation of stock-tank and nonstock-tank components does not take place on the top few trays. As the methane, etc., pass down the column, the temperature finally gets high enough to vaporize it. This material then passes up the tower as vapor until a portion is condensed by the cold feed. If this recycle is allowed to continue, the amount of vapor gets out of proportion and the column may only remove this condition by "burping"—i.e., relieving this vapor load overhead during which liquid is also lost.

For all practical purposes, the temperature is fixed at both ends of the fractionator when the feed is at constant temperature. The ratio of feed to vapor on the top tray is large, so that the temperature of the vapor leaving the tower is very dependent on the feed temperature, particularly at low pressures. It is this, which is contrary to good tower design, that accounts for burping.

Pressure sensitivity . . . It is for this reason that cold-feed towers are pressure sensitive—i.e., their separation efficiency varies markedly with pressure. The adjustment problem is simply to adjust the pressure so that most of the noncondensables are lost overhead during the initial flash without excess loss of stock-tank material.

There are almost an infinite number of pressure and temperature combinations that will work—that will give the desired product—yet the optimum range is relatively narrow. It is probably for this reason that relatively few of the field stabilizers that the author has had the opportunity to examine have been set to yield optimum recovery.

A change in feed conditions will sometimes upset the tower and cause it to become unstable. For this reason, I believe it is wise to locate the feed point several trays from the top. Much of the entrained liquid will be caught on these trays, particularly if the tower burps. The upper portion of the tower in effect acts as a scrubber for the overhead gas.

Because of this possibility of overhead liquids, some form of scrubber should always be used when the overhead gas is used as fuel.

A detailed analysis of the cold-feed stabilizer will yield the opinion that it is a relatively inflexible piece of equipment when compared to refluxed units. But it is an excellent piece of field equipment because of its simplicity

Pressure vs. recovery . . . If column pressure is plotted versus stock-tank recovery for actual field data, the curve drawn through the points would go through a maximum, the exact point depending on the feed composition and the product desired. The reasons for this are many and complex.

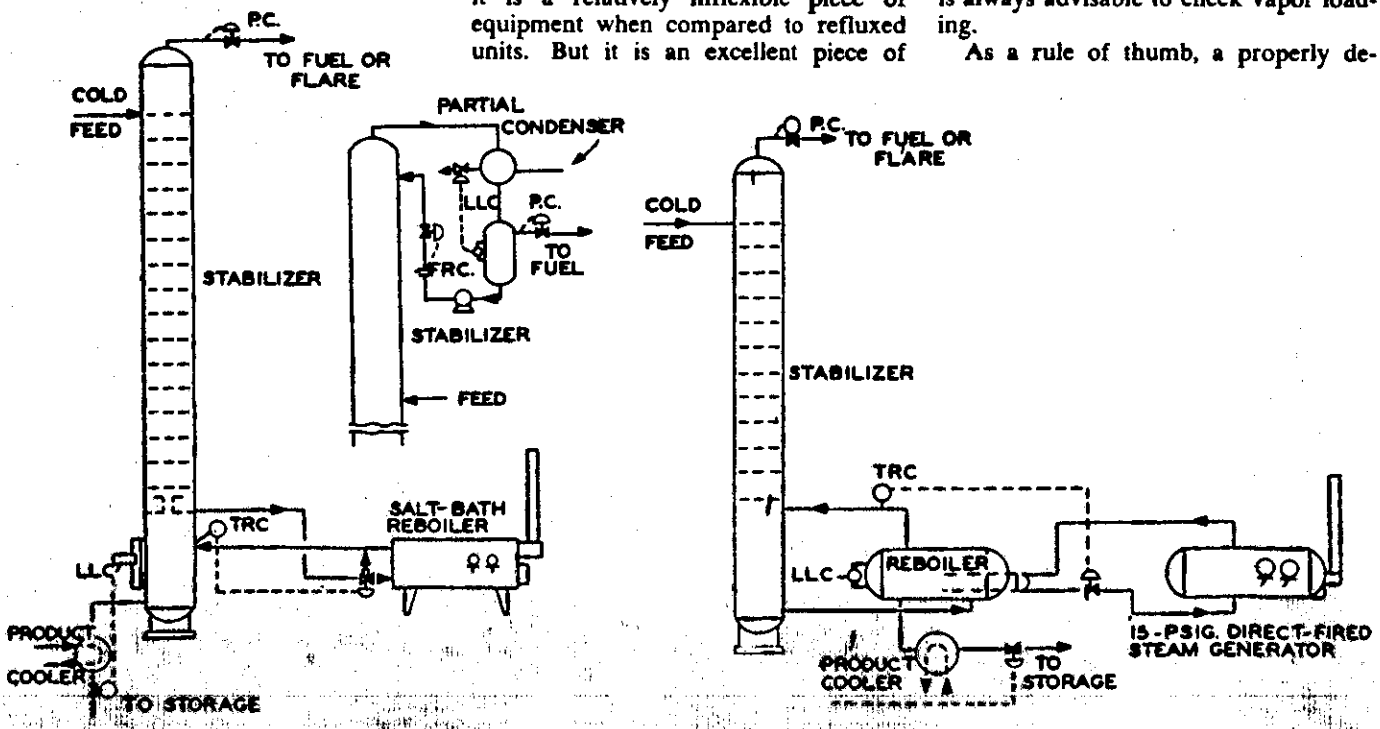
The simplest general explanation is that the initial flash loss is inversely proportional to pressure. But as the pressure is raised, higher reboiler temperatures are required, which in turn makes the overhead temperature higher. At low pressures the flash loss is controlling while at higher pressures the reboiler temperature becomes more significant.

It is partially because of the higher temperature gradient along the column that high-pressure stabilizers are generally taller. Most high-pressure towers have 20 bubble trays or the equivalent packing while low-pressure units have 10 to 12. Ten to twelve trays are enough for many stabilization jobs up to 200 psia. if the tower is carefully adjusted.

Fig. 5 shows a typical temperature-gradient curve when processing 0° F. feed in a 20-tray tower while making a stable stock-tank product.

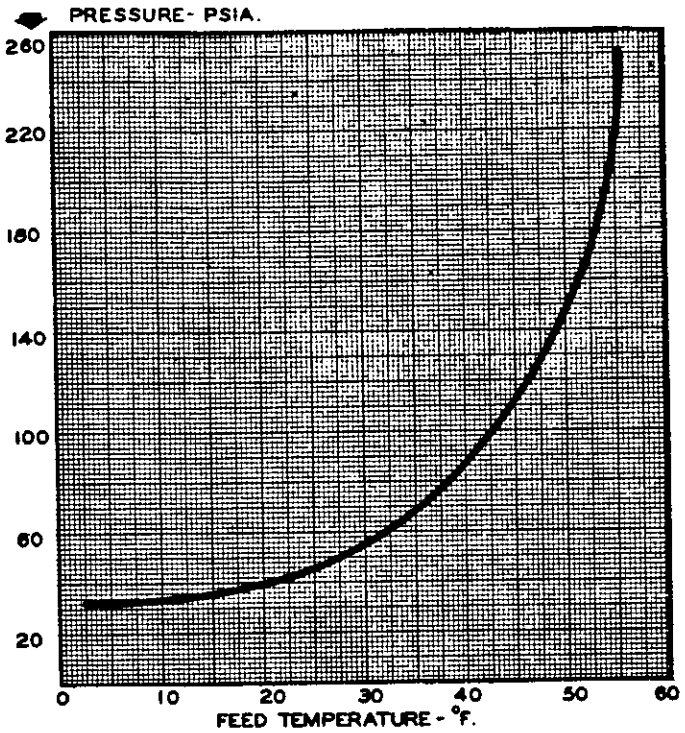
Most manufacturers use packed columns of up to 20-in. diameter and bubble tray for all larger sizes. Stabilizers are sized in accordance with established techniques, the liquid loading normally being the critical factor. However, it is always advisable to check vapor loading.

As a rule of thumb, a properly de-

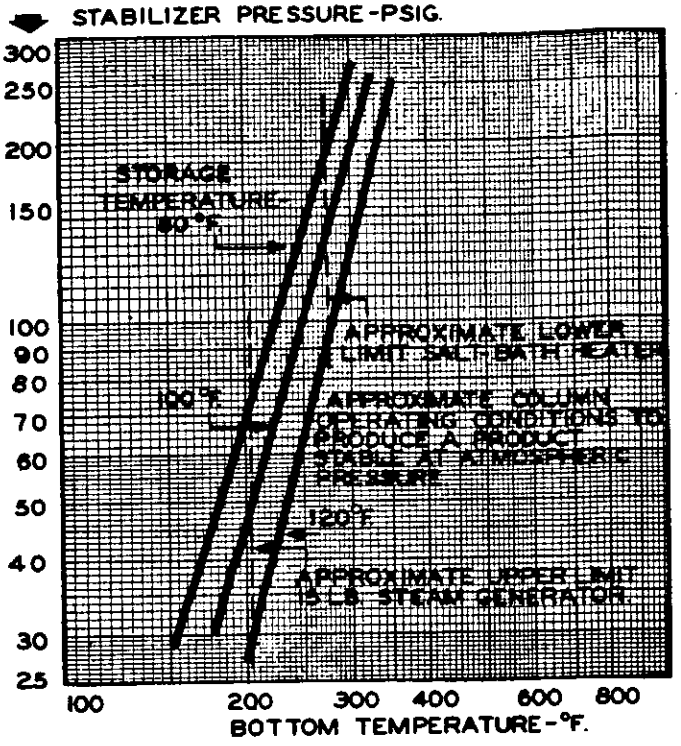


HIGH-PRESSURE cold-feed stabilizer operation. Insert shows method of handling reflux. Fig. 2.

LOW-PRESSURE cold-feed stabilizer using a gas-fired steam generator for reboiler heat. Fig. 3.



MAXIMUM FEED TEMPERATURES recommended for operation of nonrefluxed stabilizers. Fig. 4.

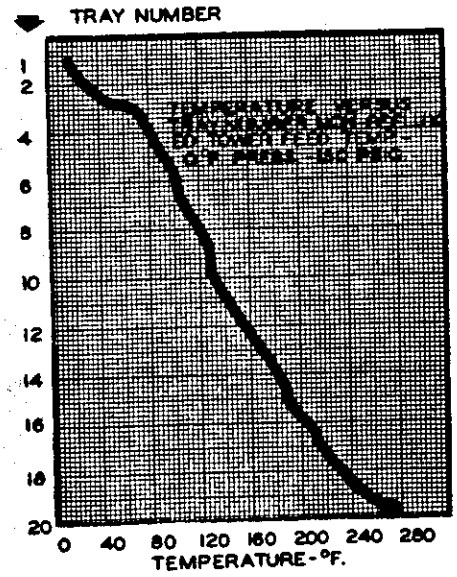


LIMITS of column pressure and reboiler temperature to stabilize a product for storage at various temperatures. Fig. 6.

signed stabilizer will handle about 100 bbl. of stock-tank liquid per square foot of column area. With a feed of 20° F. or less, over 95 per cent of the stock-tank liquid in the feed will normally be recovered in the tanks.

Cold Feed Deethanizer

Some field installations use more than one tower in order to produce propane or L.P.G., or both, in addition to oil or gasoline. Where mechanical or expansion refrigeration has been used, a cold-feed deethanizer has sometimes been specified. When used in this manner, the tower is normally operated at 350 to 400 psia. The results of this have at best been uncertain, with most of the experiences being at least partially unsatisfactory.



TEMPERATURE GRADIENT in a 20-tray high-pressure cold-feed stabilizer. Fig. 5.

The separation in a cold-feed column is a "sloppy cut" by fractionation standards although satisfactory for stabilization. Deethanization, on the other hand, requires a close cut between ethane and propane. Experience has shown that it is difficult if not impossible to get efficient deethanization in a cold-feed tower without losing excessive propane.

Table 1 is the result of an actual test on such a tower, and illustrates typical results obtained. The feed came from a low-temperature separator operating at 0° F., after passing through a surge vessel. About 14 per cent of the entering ethane is carried out in the bottoms, even though 13 per cent of the propane is lost. In this instance the quantity of propane lost may be reduced only by removing less ethane.

As a result of similar experiences, I believe that the cold-feed deethanizer only has good application in those cases where L.P.G. is being produced and when an excess quantity of propane is present (lost anyway). Otherwise it is necessary to add butane to the "commercial propane" to compensate for the surplus ethane if the column is operated to reduce propane loss. Such a fluid meets only the vapor-pressure standards of true commercial propane.

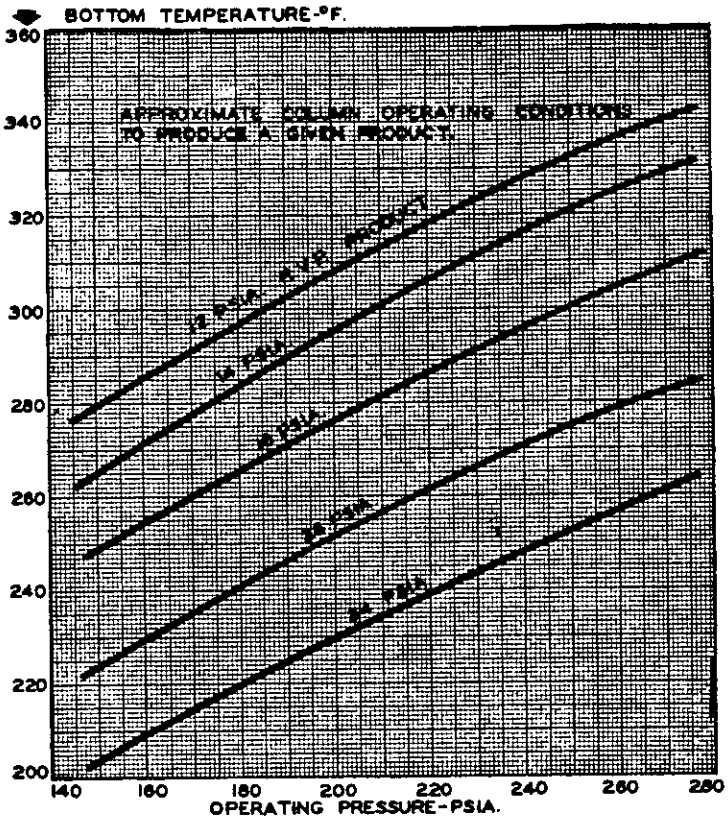
Reboiler Design

The reboiler normally used with the low-pressure stabilizer is a 15-psig.-working-pressure, gas-fired steam generator. Since steam at this pressure has a saturation temperature of 250° F., a reboiler temperature of 200° F. represents a practical controllable maximum when using this system.

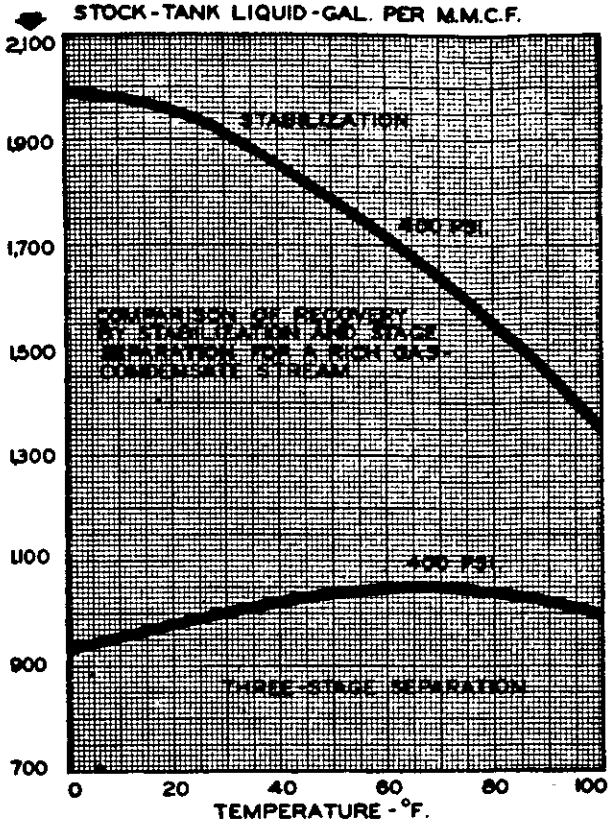
Examination of Fig. 6 shows that the column pressure may not exceed 50-70 psig. in the average application. This

TABLE 1—COLD-FEED DEETHANIZER MATERIAL BALANCE

	Feed—17,550 gal. per day		Overhead—0.224 M.M.c.f.d.		Bottoms—12,600 gal. per day	
Comp.	Mole per cent	Moles/ hour	Mole per cent	Moles/ hour	Mole per cent	Moles/ hour
C_1	17.47	11.88	48.02	11.88		
C_2	16.41	11.16	38.98	9.61	3.58	1.55
C_3	28.94	19.66	10.29	2.54	39.60	17.12
iC_4	2.91	1.98	0.62	0.15	4.22	1.83
nC_4	16.78	11.40	1.58	0.39	25.40	11.01
iC_5	3.38	2.30	0.19	0.05	5.20	2.25
nC_5	6.08	4.14	0.12	0.03	9.50	4.11
C_6	4.47	3.06	0.12	0.03	6.98	3.03
C_7+	3.56	2.42	0.08	0.02	5.54	2.40
	100.00	68.00	100.00	24.70	100.00	43.30



STABILIZER operating conditions to produce products with various vapor pressures. Fig. 7.

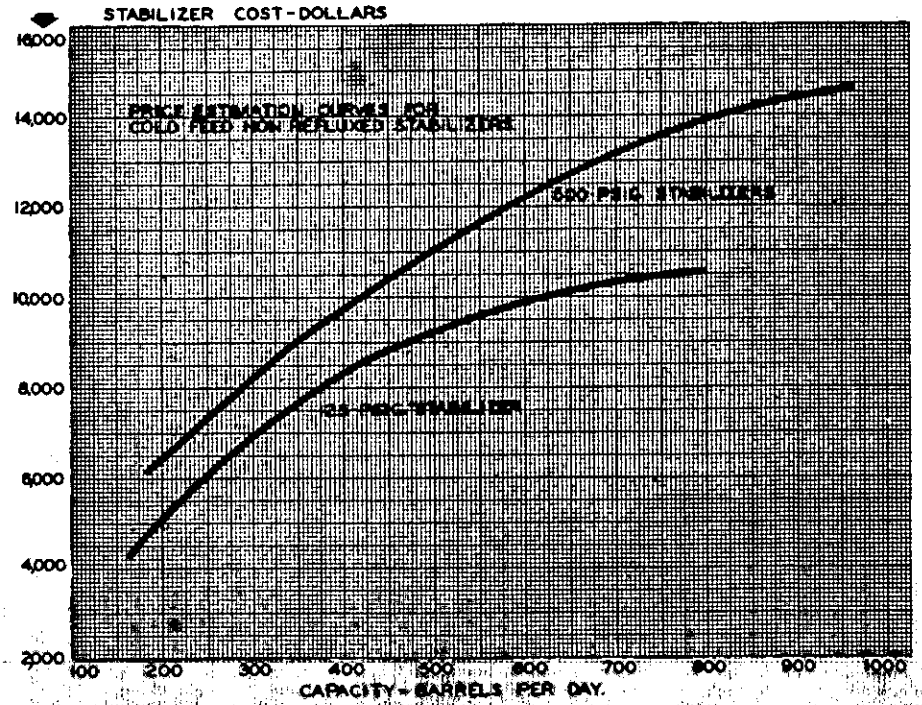


ADVANTAGE of using stabilization on a rich gas-condensate stream. Fig. 9.

obviously limits the flexibility of the unit.

A pressure of more than 15 psig. may not be used without bringing the steam generator under the boiler ordinances of the states. In most states, such boilers must be attended 24 hours a day, which is obviously impractical in the average lease situation.

The reboiler shown in Fig. 3 may be external, as shown, or be mounted in the column proper. If the latter is done the column is usually larger in diameter at that point. In either event, the reboiler must have enough elevation to allow the steam condensate to return to the steam generator if a condensate pump is not used.



COSTS of cold-feed nonrefluxed stabilizers for high and low operating-pressure ranges (as of May 1955). Fig. 8.

Temperature control . . . Effective and accurate control of temperature is obtained with the temperature-control valve in either the condensate or vapor line.

The reboiler for the higher-pressure stabilizer is usually a salt-bath heater, which is nothing more than a regular oil-field indirect heater modified for use with salt. This eutectic salt, previously mentioned, is composed of 40 per cent sodium nitrite, 7 per cent sodium nitrate, and 53 per cent potassium nitrate. It becomes molten at about 288° F. It is stable and substantially nonvolatile up to 850° F., and therefore may be used with safety in this application.

The primary disadvantage of this system is the control problem presented. The salt must be maintained at a substantially constant temperature because it is impractical to control the heat input by varying salt temperature because of the large inert mass involved. Consequently, the heat input to the tower is varied by bypassing a portion of the liquid around the reboiler. Experience has shown that best control results when a three-way valve is used close to the heater. This allows better mixing of the fluids before reentering the tower.

The drawbacks to this system are obvious, but for the most part it is very satisfactory for stabilization. It should be used with caution, though, for more exacting fractionation.

The problem of time lag with this control system has been partially offset by using controllers with an accelerated response. (Control problems such as this will be discussed in more detail in a part to follow.)

Reboiler systems using Dowtherm, Paracymene, high-pressure steam, etc., are not commonly seen in field stabilization systems to date.

Proper reboiler temperature . . . Figs. 6 and 7 are convenient to use as a guide in estimating the proper reboiler temperature for a given column pressure. Fig. 6 gives the approximate temperature needed to obtain a product stable in atmospheric storage at three different storage temperatures. The vertical dashed lines show the rough limits of each of the reboiler systems. The lower limit on the salt-bath heater is predicated on having a 50° F. differential between the salt temperature and the tower bottoms.

Obviously, a change in this differential will affect the limiting temperature. The salt bath should not be used below about 325° F. to insure that all is molten.

Between 200° and 270° F., an indirect heater using heat-transfer oil may be used. However, when making this product, the tower pressure is usually held within the temperature range of the 15-lb. steam generator or the salt-bath heater.

Fig. 7 is used when a product with a given Reid vapor pressure at 100° F. is desired. The 14-psi.-R.v.p. line roughly coincides with the 100° F. line in Fig. 6.

The actual temperature needed is the bubble-point temperature of the bottom product at column pressure. Since this is a function of composition, it is not possible to always predict the temperature accurately using the type of curve shown. Experience has shown, though, that these curves are convenient for making preliminary settings.

Stabilizer Cost

Although stabilization is superior to stage separation from a process standpoint, its use must be justified by economics. Fig. 8 is convenient for estimating the cost of stabilization equipment, including all necessary accessories. About 20 to 30 per cent should be added to the costs shown to cover transportation and installation. Stage-separation prices may be estimated by reference to Parts 1 and 5 of this series.

It is often difficult to justify the additional cost of a high-pressure stabilizer on the basis of concrete facts because it is difficult to differentiate between the recoveries to be obtained with the two models. On the basis of experience, though, I usually recommend the high-pressure model because of its greater flexibility, adaptability, and generally higher recovery. In the final tabulation of over-all well cost, the additional cost here has little effect.

Applications . . . It is difficult to accurately predetermine those places where a stabilizer will show significant advantage over stage separation. Stabilization will, however, generally show advantage in any of the following circumstances:

1. First-stage separation temperature is between 0° and 40° F.
2. First-stage separation pressure is greater than 1,200 psig.
3. Stock-tank liquid gravity is greater than 45° A.P.I.
4. It is known that the oil contains large quantities of pentanes plus, even though the gravity is not abnormal.
5. A product with controlled specifications is desired (ethane-free, propane-free, etc.).

Fig. 9 illustrates the advantage of using stabilization on the liquid separated from a gas-condensate system at 400 psig. The advantage is particularly pronounced on rich gas-condensate systems of this type.

Although the benefits are more pronounced with such systems, stabilization of crude oil often proves advantageous. It is expected that more widespread use will be made of this medium in the future.

The gas passing overhead from the stabilizer may be used as fuel for the reboiler and any adjacent gas-fired equipment. When processing a liquid separated at 0°-20° F., the gas overhead is normally 5 to 8 per cent of the total well stream, both expressed in standard cubic feet.

ELEMENTS OF FIELD PROCESSING—7

How to Sample Separator Fluids

by Dr. John M. Campbell

THE proper determination of separator conditions is dependent on a well-stream feed analysis being available. This in turn means that the said stream must be sampled and analyzed prior to this approach. The final solution reached may therefore be no better than the accuracy of sampling and analyzing. In the average case no effective control is held over the analysis except to assure oneself that the analyzer is competent.

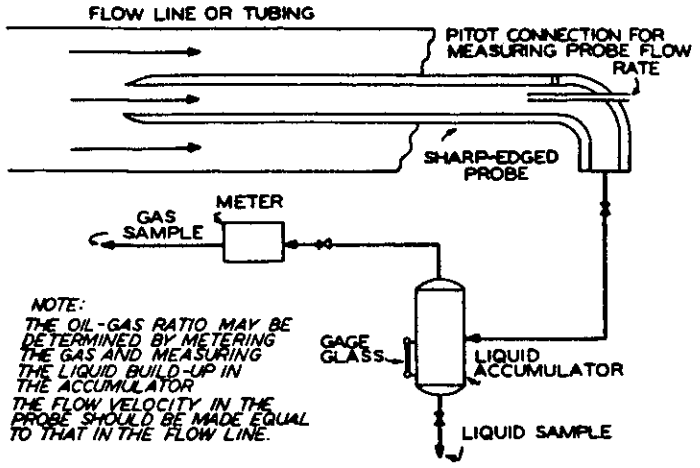
Sampling is a simple process mechanically and therein lies the danger. The average feeling is that "any fool can grab a sample." This is true but what is needed is a **representative** sample. Consequently sampling is fraught with pitfalls, the chief of which are:

1. Use of dirty cylinders that contain portions of past fluids analyzed.
2. Sampling at the wrong point to get a representative sample.
3. Obtaining sample when temperature and pressure conditions are not stable.
4. Taking the sample in such a manner that the pressure in the sampling cylinder is not the same as that in the separator.
5. Dilution of a gas sample with air.
6. Loss of sampling cylinder pressure between the time of sampling and analysis.
7. Failure to record all necessary data and properly mark the sample cylinders.

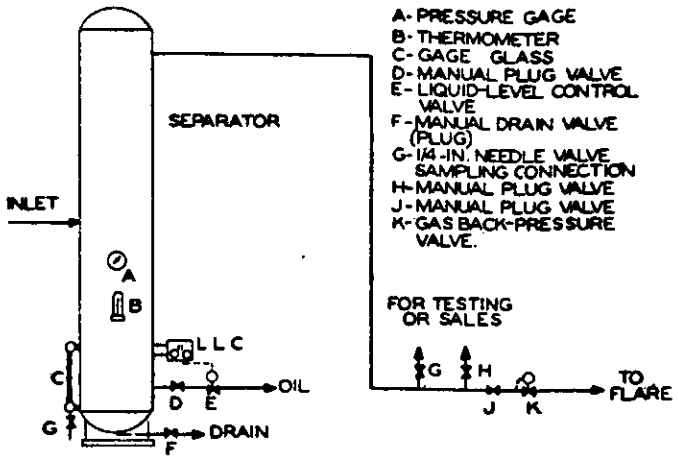
Any or all of these may introduce serious error. Many embarrassing and costly errors have resulted from improper sampling.

Even under ideal sampling conditions it must be recognized that well-stream analyses are constantly varying over short periods of time, and gradually vary, of course, during reservoir depletion.

How it's done . . . Most well streams, before separation, consist of a mixture of gas and liquid in varying proportions. This may vary from a dry-gas



PROBE-SAMPLING a gas-liquid line by a commonly used method. Fig. 1.



RECOMMENDED SEPARATOR HOOKUP for average field installation. Fig. 2.

stream containing virtually no liquid to a crude-oil well with a very low gas-oil ratio. It is possible to make an analysis of these mixed streams at a point between the wellhead and the first separator. Fig. 1 shows one method used.

A knife-edged probe is inserted into the well stream, usually at the center line, of such size and adjusted so that the velocity in the probe is equal to that in the pipe. Consequently it is expected that over a reasonable period of time a representative sample will be obtained. When carefully done this method is satisfactory but does have the following drawbacks:

1. When only small amounts of liquid are present or the velocity is low, or both, the liquid tends to flow along the walls.

2. Many wells, particularly gas-condensate, tend to produce liquid in surges so that a relatively long sampling period is needed. During such surges it is difficult to obtain representative samples.

3. Good equipment, and a trained man to operate it, are needed for good results.

This approach is excellent for obtaining well-stream gas-oil ratios for the gas sampled may be easily metered while the liquid is accumulated and measured.

In average case it is easier and more convenient to obtain the samples at the first separator battery. If proper care is used sampling at this point gives accurate results.

Separator hookup . . . The hookup shown in Fig. 2, or its equivalent, is a good one since it offers flexibility. It not only facilitates sampling but eliminates the need for repiping as the producing conditions change.

Sampling locations are provided on the gage glass and on the overhead gas line. The plug valve D is provided to have a positive shutoff ahead of the liquid level valve E, the drain also being likewise equipped (valve F). Except for valves D, F, and G, all accessories shown are standard with separator manufacturers.

The method of handling the overhead gas is recommended whether the gas is being flared or sold. Valve J may be omitted, if desired, although it is handy in the event the back-pressure valve fails to function.

If the gas is being flared connection H may be used to test the gas volume by closing off valves J or K, or both, and opening H. An orifice well tester or critical flow prover may be simply screwed into the outlet of valve H.

If the gas is being sold through connection H the back-pressure valve will still keep the separator pressure from going too high by flaring any excess gas automatically. In such cases it is often wise to also have a back-pressure controller in the sales line to keep from drawing down the separator pressure. This is particularly true when a group of low-pressure separators are tied together on the suction side of a compressor station.

Separator Sampling

Sampling a stream at a separator is the recommended procedure for it provides control over all the factors. When sampling in this manner it is necessary to determine the gas-oil ratio in the separator as well as obtaining the liquid and gas samples.

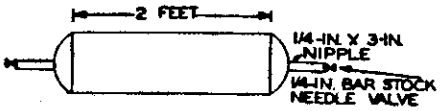
The gas sample should be taken close to the separator but after the gas has left the separator proper. **Under no circumstances** should it be drawn from the separator shell, for the wells of that vessel are covered with liquid which will contaminate the sample. A very small amount of liquid will cause an appreciable error in a gas sample. For these reasons the gas sample is normally obtained from a meter run (if present) or from connections G and H.

When no such connections are available it is sometimes absolutely necessary to obtain the gas sample from the pressure-gage connection A in the shell of the separator. In this case it is necessary to improvise a method of scrubbing out any entrained liquid. On low-pressure separators an instrument gas drip pot may be used. A length of pipe filled with desiccant is also good. Some similar piece of equipment is a recommended part of the sampler's kit of tools.

The liquid samples may be normally withdrawn from the separator gage glass. Installation of a ¼-in. needle valve (G) is recommended for this purpose.

Sample Bomb Construction

All sample bombs used, regardless of the pressure rating, should be cylinders with valves on both ends. The ends should be closed either by (a) swages reducing to ¼-in. diameter, threaded ¼-in., or (b) by pipe caps which have been drilled in the center to accommodate ¼-in. nipples, welded to the caps.



The above figure outlines the essential details of a "standard" sample bomb. The following sizes are generally used:

Service—	Diameter (in. o.d.)	Pressure range (psig.)
Gas	2⅜	Above 225
Gas	3½	90-225
Gas*	12	15- 90
Gas*	6	30-400
Liquid	2⅜	(†)
Gas	(‡)	0- 15

*These are war surplus stainless steel oxygen cylinders now in common use throughout the industry. †0 to working pressure of bomb. ‡5 gal. water bottle.

All fabricated bombs should be made of extra heavy seamless pipe, using ¼ by 3-in. extra-heavy nipples and bar-stock needle valves. The working pressure should be stamped on the barrel of the bomb. In every case the bomb should be hydrostatically tested to 1½ times the stamped working pressure.

It is advisable that a given sample bomb not be used for sampling both gas and liquid, to prevent undue contamination. To avoid mixup, gas sample bombs should be painted aluminum between the seams with the ends painted red. They should have the following stenciled in red on the shell:

Gas: Max. W.P. = xxxx Psig.

Liquid bombs should be painted red only with the following stenciled in aluminum on the shell:

Liquid: Max. W.P. = xxxx Psig.

It is desirable that crates be made for shipment of sample bombs. Full samples should be shipped express wherever possible and the agent should be advised that a red label shipment is being made consisting of oil or gas samples under pressure.

Sample bomb preparation . . . Prior to taking samples all sample bombs should be properly cleaned. They should be placed in a vertical position with the bottom valve opened to drain all free liquid. They should then be thoroughly steamed out or washed with kerosine. If the latter is used the washing should continue until no color is shown on draining. Following a kerosine rinse, fresh clean water should be flowed upward through the cylinder and allowed to overflow out the top until the fluorescent oil film disappears.

If steam is used the cleaning should continue until the effluent from the bottom is clear and without fluorescence. The cleaning of the bomb is an integral part of sampling and should always be done personally by the sampler wherever possible.

Sampling

In order to obtain a representative sample the fluid in the bomb at the end of sampling must be at the same conditions as the sampling. Air contamination must also be prevented. Consequently it is recommended that the bombs be filled with water prior to sampling, the sample being taken by water displacement. In some instances bombs are evacuated prior to sampling, and this is satisfactory if one can be assured that no air leakage has occurred. The use of water increases the weight of the bomb but eliminates this possible error.

If possible the bomb should be filled with water just prior to sampling for water expands on warming up and the bombs could become overpressured. In any event they should be stored in a cool place.

Gas . . . If the sample is being taken from an orifice-meter manifold, bypass the meter and connect a length of copper tubing to the bleeder valve. Connect the other end to a tubing connection made up in the top valve of the sample bomb but leave only hand tight for the moment. The bomb should be placed in a vertical position and the following procedure used:

1. Open the bleeder valve slightly, allowing enough gas to flow through the tubing to purge the air it contains.
2. Tighten the tubing nut with a wrench and allow the pressure in the tubing to build up to line pressure.
3. Open the bleeder valve and the top valve on the bomb completely and tighten the packing nuts if gas leaks are observed.
4. Crack the bottom valve on the sample bomb and allow the water to bleed off SLOWLY. Take at least 5 minutes to discharge the water.
5. After the water is discharged, let gas continue to flow through the bottom valve at the same rate for at least 15 or 20 minutes more.
6. DO NOT LET GAS EXPAND INTO THE BOMB THROUGH THE TOP VALVE. ALL EXPANSION MUST BE TO ATMOSPHERE VIA THE BOTTOM VALVE.
7. Close bottom valve, close top valve, and close bleeder valve in the precise order named. Disconnect tubing, dope pipe plugs with soap, and make up tight in both bomb valves. **Tighten valve packing nuts!**
8. Head for the nearest water tank, slush pit, or puddle, immerce the whole bomb and check CAREFULLY for leaks. Take your time with this check and look carefully for minute bubbles.

If the gas pressure is low and a 5-gal. water bottle is to be used to take an atmospheric-pressure sample, the following procedure applies. Equipment needed includes a rubber stopper, or a wax-impregnated cork, to fit the bottle; a fairly long piece of rubber or copper tubing; a 10-qt. pail or a wash tub, and a supply of clear, fresh water. Proceed as follows:

1. Fill the bucket or tub full of water and set it adjacent to the sample point. Fill the bottle with water, place palm of hand over opening and invert the bottle immersing neck below water level in bucket or tub. IF AIR BUBBLES APPEAR, TRY IT AGAIN SO THAT THERE IS NO AIR TRAPPED IN THE BOTTLE.
2. Connect tubing to sample valve, crack valve to purge air from the tubing, then pass the free end of the tubing under the water level and up inside the bottle. Gas will then displace the water which will overflow the bucket or tub.
3. Note the time required to displace the water from the bottle and allow gas to pass into the bottle for a period four times as long. This is necessary to eliminate the effects of absorption of carbon dioxide by the water.
4. Remove tubing and insert stopper in the bottle while the neck is still immersed. Make sure stopper is tight, then remove bottle from tub.
5. Wire stopper securely to bottle neck and, if possible, dry stopper and glass neck surfaces and coat with wax. Place in crate for shipment.

IMPORTANT SOUR-GAS NOTE! Always take sour-gas samples in a glass bottle unless special bombs are available. H_2S reacts with steel making analysis impossible. Obviously, these samples will be at atmospheric pressure.

Liquid . . . Liquid samples should be taken from the liquid-level gage glass according to the following procedure:

1. Close gage-glass cocks, bleed off pressure, replace drain cock with suitable nipple or swage and a ¼-in. bar-stock valve, if necessary. Close drain valve, open both cocks, and bleed liquid from the glass slowly via the drain valve to purge.
2. Connect copper tubing to drain valve and make up other end of tubing hand tight into top valve of sample bomb. Crack drain valve and allow liquid to purge air from tubing.
3. Gradually draw up tubing nut until it is tight. Turn drain valve and top valve on bomb wide open.
4. CLOSE THE TOP GAGE COCK.
5. Crack bottom bomb valve and withdraw the equivalent of a full pint of water. **Do this slowly, making sure a liquid level is always in sight in the gage glass.**
6. Close the bottom bomb valve. CLOSE THE BOTTOM GAGE COCK. OPEN THE TOP GAGE COCK.
7. Crack bottom bomb valve and SLOWLY withdraw the remainder of the water.
8. Close the bottom, top valves of the bomb, and the drain valve on the gage glass in the order named. Disconnect tubing. Dope pipe plug threads with soap and pull up tight in valves. TIGHTEN VALVE-PACKING NUTS
9. Test carefully for leaks, under water, as discussed before.
10. Return gage glass on separator to normal service.

The above procedure is preferred above all others since it provides sufficient sample at system pressure and

temperature for accurate analysis, vastly simplifies laboratory procedure, and insures safety in shipment by providing a substantial gas cap. Be sure to draw water before taking the sample and follow this procedure carefully.

With special reference to Steps 6 and 7 in the above procedure, make sure that the liquid level disappears when the bottom gage cock is closed and the top cock opened. Failure to dissipate the liquid level means the bottom cock leaks and the bomb will be filled completely with liquid. This is wholly undesirable, so shut the bottom cock TIGHT!

When possible the liquid and gas samples should be withdrawn simultaneously.

Gas-Oil Ratio

When taking gas and liquid equilibrium samples from a separator it is also necessary that the gas-oil ratio be obtained. Knowing this, the separator pressure, and temperature, these analyses may be recombined to yield a composite well stream.

In the ideal case where the residue gas is being sold the gas-flow rate is simply obtained from the meter-run data. In the absence of the meter run gas volumes may be determined using a critical-flow prover, orifice well tester, or pitot tube, using the test connection shown in Fig. 2.

Whenever the separator pressure is above 15 psig. the critical-flow prover is preferred because of its greater accuracy. The orifice well tester is relatively inaccurate as is the pitot tube unless very carefully used.

Except in those relatively rare instances where a metering separator is available the liquid flow rate from a vertical separator is most easily determined by measuring the rate of liquid rise in the separator. If Valve D has been included (Fig. 2) this is easily done by:

1. Making two marks on the gage glass at least several inches apart.
2. Manually reduce the liquid level below the lower mark.
3. Close Valve D and/or close the liquid-level control valve.
4. Start a stop watch when the liquid level reaches the lower mark.
5. Record the time necessary for the liquid to reach the second mark and the distance between the marks.

The time elapsed should be several minutes for good accuracy. The distance of rise may be converted into gallons (or barrels) of liquid by consulting a sales drawing of the separator manufacturer. If any appreciable quantity of water is being produced with the oil the volume obtained above should be corrected. The rate of water production may be easily determined by using drain F and a container of known volume.

Whenever possible the incremental length used for the liquid determination should fall below the float flange (if any) in order to eliminate the error it might introduce by changing the separator's cross-sectional area.

In a horizontal separator the variation in volume with separator height is not linear, requiring a separate calculation. Once again after reference to the manufacturer's drawings, a calibration curve may be calculated for a given separator.

If not enough information is available for the calculation of volume the only alternative is to shut down the separator, depressure, drain of oil, and calibrate using fresh water. It is actually a good practice to calibrate a separator in this manner at the time of installation so that the data are available.

SUMMARY OF SEPARATOR TESTS

Company: _____

Well and Lease: _____ Field: _____

County: _____ State: _____ Date: _____

Gas Metered With: _____

Base Temperature: _____ °F. Pressure: _____ Psig.

Barometric Pressure: _____ in Hg. Atmospheric Temp.: _____ °F.

DATA	Run No. 1	2	3
Meter Run Diameter - inches			
Orifice Diameter - inches			
Static Pressure - psig.			
Static Pressure - H_2O			
Differential Press. - H_2O			
Gas Temperature - °F.			
Specific Gravity Gas (Air = 1.00)			
Corrected Orifice Coefficient (hr.)			
Daily Flow Rate - M.c.f.d.			
Diff. Volume Stock Tank - Bbl.			
Time Interval - Minutes			
Diff. Volume Separator - Bbl.			
Time Interval - Minutes			
Liquid Rate, Bbl. per Day			
Gas-Oil Ratio			
Specific Gravity Liquid - °A.P.I.			
Separator Pressure - psig.			

CLEVELAND ENGINEERING COMPANY

By: _____

SEPARATOR CALIBRATION—This form is used to summarize the tests made in calibrating a separator. Fig. 3.

A data sheet of the type shown in Fig. 3 is handy for the recording and calculation of test data. Once the samples are obtained they should be tagged, preferably with linen tags, and the following information included:

Description of sample
Location of separator
Date, time
Separator pressure
Separator temperature
Atmospheric temperature

If all of these standards are adhered to, the representative samples obtained will help immeasurably in the choice of proper operating conditions.

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ELEMENTS OF FIELD PROCESSING—8

How Liquid Type Dust Scrubbers Clean Gas

by John M. Campbell

SEPARATION of dust and entrained liquid is primarily a pipeline rather than a lease processing problem, but merits discussion since it is partially an outgrowth of increased processing at the lease. The problem has become greater as the quantity of gas handled has become larger. The presence of these contaminants and their effect on pipeline efficiency must be considered as one of the main problems in gas transportation.

Efficient cleaning of gas is hampered by the widely varying properties of the contaminants and the lack of classification standards to facilitate engineering design calculations. Consequently many problems are handled by trial and error, rules of thumb, and the rather nebulous factor, "experience." These last two are often found lacking, with the result that expensive trial and error offers the only recourse.

Equipment now being used may be subdivided into two broad classifications—that which uses liquid as the primary scrubber and that which does not. Equipment in the latter category will be discussed in Parts 9, 10, and 11 to follow.

Characteristics of liquid scrubbers . . . All liquid scrubbers are simply vessels so designed to provide intimate contact between the gas to be cleaned and the scrubbing liquid. The general design shown in Fig. 1 is typical of the vertical type.

Except for the lower section the general features are similar to those of oil and gas separators, a primary separation and mist extraction section being provided. Many different internal arrangements have been used in the scrubbing section, but the most common has the gas flow through the liquid. Their contact is aided by some sort of contact device. This liquid is then removed in the separation section and returned by gravity to the liquid sump.

Other arrangements are used but they usually involve the use of an external power source. Fig. 2 shows a horizontal vessel divided into two sections. The first acts primarily as a liquid knockout while the other is the liquid scrubber. The latter consists of corrugated cylinders which are rotated by an external motor.

The lower portion of the vessel is filled with oil and as the cylinders rotate they become coated with oil. Dust passing through the oil labyrinth formed by the cylinders will tend to cling and is subsequently removed when the rotating cylinder passes through the oil. As expected, this unit is reported to have about three times the capacity of an equivalent vertical unit, but has the obvious disadvantage of requiring an external source of power.

Poblems of liquid carryover . . . A properly designed liquid scrubber is relatively efficient, but has the disadvantage of introducing liquid downstream. A carryover of 0.1 gal. per M.M.c.f., or less, is normal but sometimes entire charges of scrubber liquid have been lost. This carryover leads to the danger of hydrostatic lock in compressor cylinders and damage to metallic packing. Where fiber compressor valve plates are used there is often danger of softening, causing early failure. The normal carryover is, of course, limited by the efficiency of the mist extractor used.

Abnormal loss is usually due to one or more of the following:

1. Dilution of the scrubber oil by incoming liquid or absorption of components from the gas.
2. Foaming.
3. Surges of liquid due to rapidly fluctuating gas loads and pressures.
4. A combination of gas flow rate and pressure that exceeds the rated capacity of the vessel.

Eliminating carryover . . . The first problem may be minimized by putting a conventional "dry" scrubber ahead of the liquid scrubber, but this introduces obvious economic drawbacks.

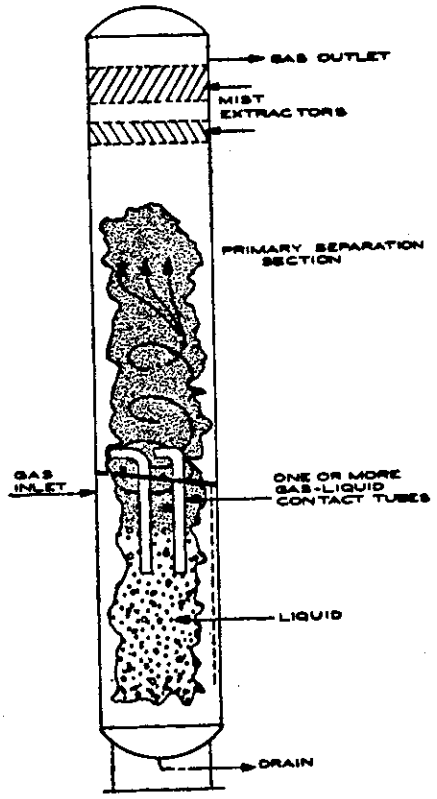


Fig. 1—Typical vertical-type liquid scrubber.

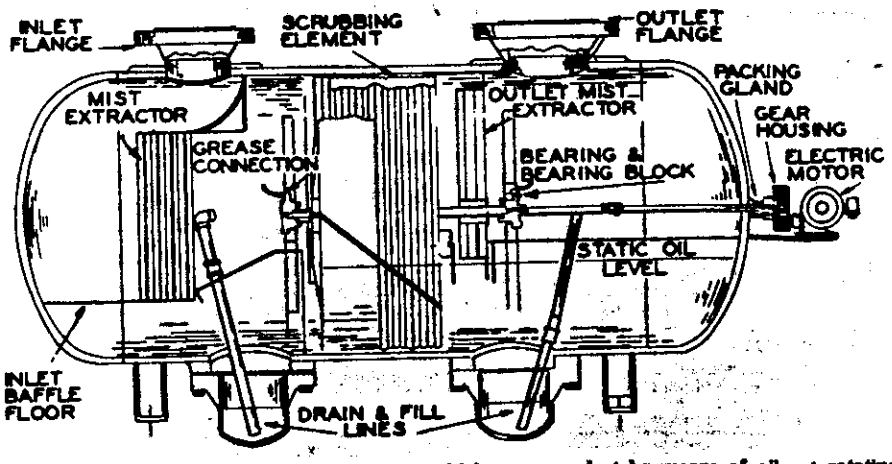


Fig. 2—A type of horizontal liquid scrubber which removes dust by means of oil-wet rotating cylinders.

Some have used a low-vapor liquid immiscible with oil such as diethylene glycol, periodically or automatically skimming off any oil collected. There are some successful installations of this type but glycol is relatively expensive. Except for very rich gas the absorption of light components is normally negligible

Foaming may result for various reasons, among which are the presence of aromatics or foreign liquids such as those injected into wells and pipelines to minimize corrosion.

Liquid surging may be largely prevented by proper design and operation of the line. Provision for bypassing the scrubber during blowdowns, etc., is recommended.

Line pressures normally vary, due partially to the widespread practice of packing lines during periods of off-peak demand. Consequently such vessels should be sized at the minimum and maximum pressure, and maximum flow rate, anticipated. One reported investigation[2] disclosed that the abnormal losses of oil were obtained at operating pressures 12 per cent over design with the flow rate 16.5 per cent above capacity. However, at an operating pressure of 37.7 per cent over design and a throughput only 2.5 per cent over design capacity the oil losses increased to about 0.51 gal. per M.M.c.f.

It is because of these carryover problems that many operators have resorted to dry-type scrubbers although there are many successful liquid units in operation.

Scrubber liquid . . . The liquid used should have a low vapor pressure, low pour point, relatively low viscosity and be able to "wet" the dust. One manufacturer[3] recommends any oil having the following general properties:

I.b.p.—500° F. minimum.
E.p.—800° F. maximum.
Gravity—25°-35° A.P.I.
Viscosity—100 S.U.s. at 100° F. on lean gas below 500 psig. 150 S.U.s. at 100° F. on rich gas and/or above 500 psig.

Most dust and debris in pipelines is efficiently scrubbed with most any liquid but two cases are known where material from the producing formation went through two separators and a glycol dehydrator containing four contact trays and then accumulated in a desulfurizer absorber. Fortunately this is the exception rather than the rule.

Available gas rate . . . The capacity of vertical dust scrubbers may be estimated by the equation:

$$Q = 5.55\,(D^2P/P_BT)\,[(\rho_L - \rho_G)/\rho_G]^{1/2} \quad (1)$$

where:

Q = flow rate—M.M.c.f.d., measured at pressure P_B and 60° F.
P_B = std. metering pressure—psia.
P = actual flowing pressure—psia.
T = actual flowing temperature—°R. (°F. + 460).
ρ_L = scrubber liquid density—lb. per cu. ft.
ρ_G = actual gas density—lb. per cu. ft. at P and T.
D = effective internal diameter available for gas flow—ft.

This equation stems from the equations commonly used to determine the allowable velocity in towers to prevent excess liquid entrainment. The coefficient used will vary somewhat with the manufacturer and the design, but the use of 5.55 will at worst be slightly conservative. For horizontal scrubbers a value of 15 may be safely used.

The last term of Equation 1 shows that the separation of liquid and vapor become more difficult as their densities become closer. The density of the liquid is substantially independent of pressure while that of the vapor is roughly proportional to pressure. Therefore, as the pressure is increased the last term becomes smaller. However, this is more than compensated for by the increase in P in the second term. Consequently Equation 1 would indicate an increase in capacity with pressure.

This is contrary to the results sometimes found in actual practice. In fact the liquid carryover problem has generally become more pronounced as general scrubber pressures have increased. It would therefore appear that the coefficient in Equation 1 also varies with pressure. The value used in Equation 1 has been found satisfactory at pressures to 1,000 psia, however. For other than estimation purposes reference should be made to guarantee capacity curves of the manufacturer.

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FIELD INSTALLATION of industrial dust scrubbers for natural-gas service. These vessels are located at a field plant of Oklahoma Natural Gas Co. near Depew, Okla. (Photograph courtesy of Maloney-Crawford Tank & Manufacturing Co.)

ELEMENTS OF FIELD PROCESSING—9

Mist Extraction—An Art, or a Science?

This is the first of several installments on mist extraction. The author presents some general principles before discussing available equipment

by John M. Campbell

MIST extraction is one of the most important factors entering into the design of field processing equipment, including separators, stabilizers, scrubbers, etc. Although the principles are well established, at the present time their design is somewhat more an art than a science, the factors used being largely empirical.

The equipment used usually employs one or more of the following mechanisms: (1) gravity settling, (2) centrifugal force, (3) impingement, (4) electrostatic precipitation, (5) sonic precipitation, (6) filtration, (7) adhesive separation, (8) adsorption, or (9) thermal.

The problem is amplified because particles of varying size and characteristics, both liquid and solid, must be removed. Furthermore size and cost of the equipment required are always practical factors.

Particle size . . . This is normally defined by its diameter in microns, 1 μ being 1/25,400 in. The particles greater than 10 μ may be readily separated by existing equipment. Those smaller, sometimes referred to as particulates, represent a serious problem. Up until the present time at least the use of settling, impingement, centrifugal force, and filtration have not been generally effective in handling these small particles. The other processes as yet have not been generally used on high-pressure gas because of several serious limitations.

There are several ways of describing the average particle diameter:

1. Diameter of a sphere having average volume:

$$Dp = \left[\frac{\Sigma (n D^3)}{\Sigma n} \right]^{1/3} \quad (1)$$

2. Diameter of sphere having average area:

$$Dp = \frac{\Sigma (n D^2)^{1/2}}{\Sigma n} \quad (2)$$

3. Diameter of sphere having geometric mean diameter:

$$Dp = c^{-B} \quad (3)$$

where:

$$B = \frac{\Sigma (n \ln D)}{\Sigma n}$$

Dp = average particle diameter
D = diameter of given size particle
n = number of particles of given size (dimensionless)
(D and Dp in consistent units)

The normal probability of particle sizes normally takes the form shown in curve a, Fig. 1. Often the curve will be skewed to the right or left as with curve b or several peaks may be encountered as with c.

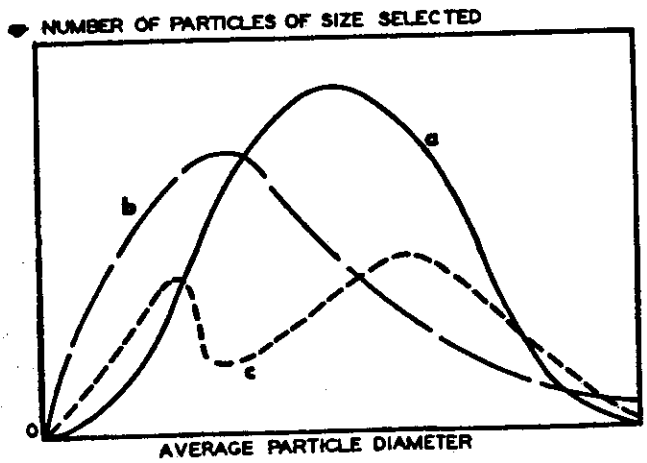
The determination of average particle size is relatively easy with solids but no method yielding reproducible data has yet been devised for liquids. This determination of particle size may be considered somewhat academic because size changes continually in an actual line—yet it is by necessity the logical starting point in mist-extractor design.

The general principles and limitations of the most common processes are reviewed here as a preface to detailed discussions on available equipment. Detailed mathematical treatment for these processes may be found in the references.

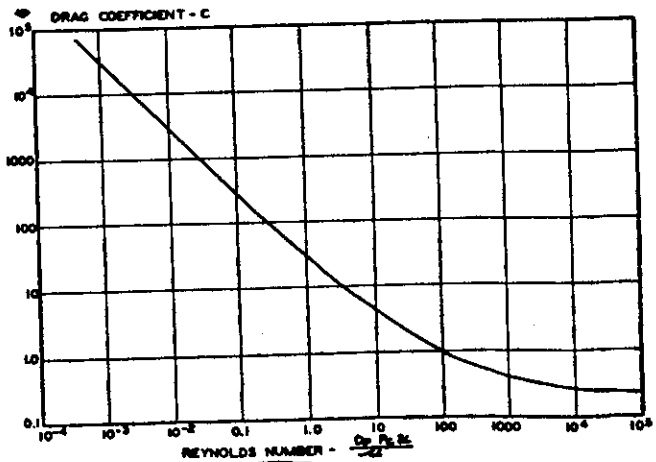
Properties of Particles

Most particles have no electrical charges of any magnitude, therefore if a charge is desired (electrostatic precipitation) this must normally be imposed by artificial means. For all practical purposes the particles may be considered inert.

Although particles of widely varying absolute density are encountered, their bulk density is generally not much larger than that of the gas. Conse-



TYPICAL particle-size distribution. Fig. 1.



VARIATION of drag coefficient C with Reynolds number—for spheres.[4]

"Determination of particle size is logical starting point in mist-extractor design."

quently separation by gravity is logically limited to particles of relatively large size. It is only above 10 μ that their higher density starts having an effect.

The vapor pressure of a small particle does not differ from that of a large mass of the same fluid until the size reaches about 0.01 μ. Since a particle of this size is not considered entrainment, it may be said that vapor pressure remains constant.

An entrained liquid system is basically unstable, the particles either coalescing or vaporizing if given sufficient time. This time is basically inversely proportional to size and proportional to the amount of interparticle contact. It is on this latter premise that impingement separators are based. Basically then the purpose of a liquid mist extractor is to accomplish this disappearance in a reasonable length of time.

Liquid properties such as surface-tension affect the coalescence and subsequent removal and therefore enter into the design.

The varying chemical properties of particles is of little consequence for the processes normally used do not involve these considerations. The difference in chemical properties between glycol and crude oil, for example, does not affect entrainment separation although their physical properties do.

General Considerations

The general physical "laws" governing the behavior of particles in an idealized system have been formulated (see bibliography). In their application to actual systems, however, certain assumptions are necessary which limit their effectiveness. They provide though a means of measuring the relative effect of various influences on particle behavior. In reality many commercial mist-extraction units combine several of the mechanisms listed above and it is difficult, if not impossible, to determine their absolute effect, or their effect on each other. However, from a mathematical standpoint, it is necessary to consider each process separately.

Gravity separation . . . The terminal velocity of a particle settling under the action of gravity may be found from the equation:

$$v_t = \left[\frac{4g\,Dp\,(\rho_L - \rho_G)}{3\,\rho_G\,c}\right]^{1/2} \quad (4)$$

which takes into account the gravitational forces and those resisting settling. Newton's law describes the resisting or drag force resisting the motion of the particle during turbulent flow. This force is normally described by a drag coefficient g. The relationship between this coefficient and Reynolds number is shown in Fig. 2. In the turbulent-flow region ($500 < Re < 200{,}000$) Fig. 2 shows that c has an average value of 0.44 for spheres.

If the gas flow is laminar (viscous) the relationship developed by Stokes applies and Equation 4 becomes:

$$v_t = \frac{g\,Dp^2\,(\rho_L - \rho_G)}{18\,\mu} \quad (5)$$

Centrifugal settling . . . Stokes law may be applied to this process if the effect of gravity g is replaced by a, the acceleration due to centrifugal force. The terminal velocity then becomes:

$$v_t = \frac{a\,Dp^2\,(\rho_L - \rho_G)}{18\,\mu} \quad (6)$$

By considering the gas velocity to be constant it is possible to obtain:

$$Dp = \left[\frac{9\,\mu\,s}{\pi\,V_o\,N\,(\rho_L - \rho_G)}\right]^{1/2} \quad (7)$$

where:

Dp = particle size that will be collected by centrifugal force.

Impingement . . . As the gas approaches a fiber it curves around but the momentum of the entrained droplet tends to move it straight ahead. It therefore becomes critical to determine the distance across the flow path necessary to stop the particle. On the assumption that Stokes law applies:

$$s = \frac{Dp^2\,\rho_G\,V_i}{18\,\mu} \quad (8)$$

These equations and others describing other mechanisms of separation are shown by Montross[7] and Lapple.[5]

Nomenclature

- a = acceleration due to centrifugal force, ft./sec.^{2}
- c = drag coefficient, dimensionless.
- D = any particle diameter, ft.
- Dp = average particle diameter, ft.
- g = acceleration due to gravity, ft./sec.^{2}
- n = number of particles of given size, dimensionless.
- N = "effective" number of turns made in centrifugal separator.
- v_t = particle terminal velocity, ft./sec.
- v = relative velocity between particle and main body of gas, ft./sec.
- V_o = average velocity of particle laden gas, ft./sec.
- V_i = initial velocity of particle, ft./sec.
- ρ_L = density of liquid, lb./ft.^{3}
- ρ_G = density of gas, lb./ft.^{3}
- μ = viscosity of gas, lb./ft./sec.
- s = radial path of particle, ft.
- π = 3.1416.

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ELEMENTS OF FIELD PROCESSING—10

Vane-Type Impingement Mist Extractors

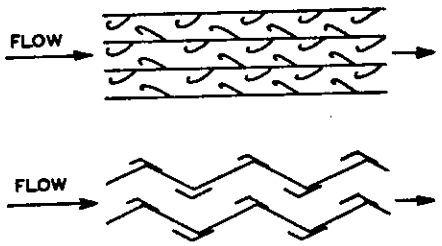
by John M. Campbell

MOST production equipment involving the separation of oil and gas at present uses impingement-type mist-extraction elements. This element is usually of the vane type or of knitted wire, the former being shown in Fig. 1.

The vane type consists of a labrynth formed with parallel metal sheets with suitable liquid collection "pockets." The gas, in passing between plates, is agitated and has to change direction a number of times. Obviously some degree of centrifugation is in rocuced, for, as the gas changes direction, the heavier particles tend to be thrown to the outside and are caught in the pockets provided.

How mist is removed . . . Coalescence of small particles into those large enough to settle by gravity is provided by two mechanisms—agitation and surface. The surface of the element is usually wet and small particles striking same are absorbed. Inasmuch as the pockets are perpendicular to the gas flow, the liquid thus formed does not have to flow against the gas. Consequently, small compact units have a large capacity.

Although such units are generally very similar they differ in construction details, such as the two shown in Fig. 1. The design is largely empirical, and no correlations apply. Generally though, the manufacturers of such units guarantee the entrainment not to exceed 0.1 gal. per M.M.c.f. when incorporated into the top of a properly designed oil-gas separator.



TYPICAL vane-type mist extractors. Fig. 1.

Designing for efficiency . . . As the plates are placed closer together and more pockets are provided, greater agitation, centrifugal force, and collection surface are provided but the pressure drop is increased correspondingly. Thus, for a given flow rate, the collection efficiency is normally some function of the pressure drop.

In the average application this pressure drop varies from 1 to 10 in. of water. Because of this pressure drop and to prevent gas bypassing the extractor, a liquid-collection pan incorporating a liquid seal is necessary for the liquid to drain properly.

This type of mist extractor is only generally efficient where the particle diameter exceeds 10 μ. When small solid particles are encountered in the absence of liquid, the efficiency is generally low. Consequently, this type is regarded as primarily suitable for the separation of liquid.

Properties of liquid important . . . The density of the particle to be separated is an obvious variable. In addition, surface tension must be considered. For example, it has been necessary to use two mist extractors in series in glycol units since that material has a high surface tension. A portion of the glycol film "flows" over the metal surface propelled by the gas velocity. A portion of this film is then reentrained, to be caught in the second unit. In this case economics does not justify more than two units in series.

A second problem has been encountered where oil or condensate is brought into the contactor along with the gas. This liquid reduces both the density and surface tension of the entrained liquid and tends to increase losses. One solution has been to add a high density anti-foam agent to the glycol to reduce this loss. In one installation the loss was reduced from 29 to less than 0.5 lb. per M.M.c.f. by this means[1].

The above examples illustrate the effect of particle characteristics on mist extractor performance.

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ELEMENTS OF FIELD PROCESSING—11

Knitted-Wire Mist Extractors—and How They Work

They increase efficiency at low installation cost

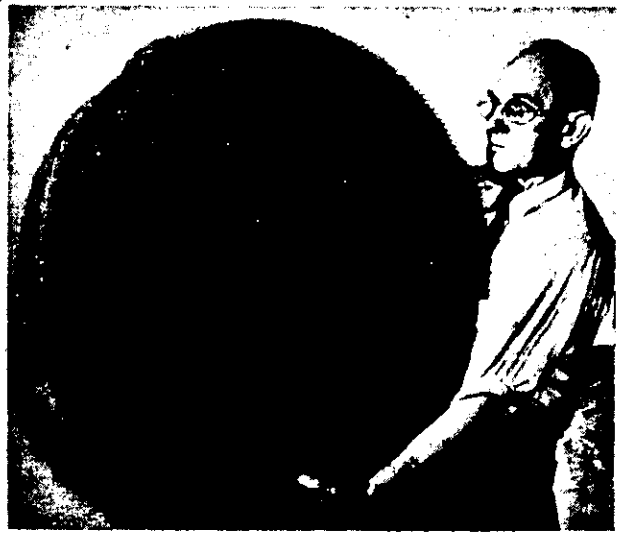
by Dr. John M. Campbell

IN the last several years, increased use has been made of mist extractors composed of a knitted wire mesh supported on a lightweight support. This material has given generally favorable results and has a low installed cost.

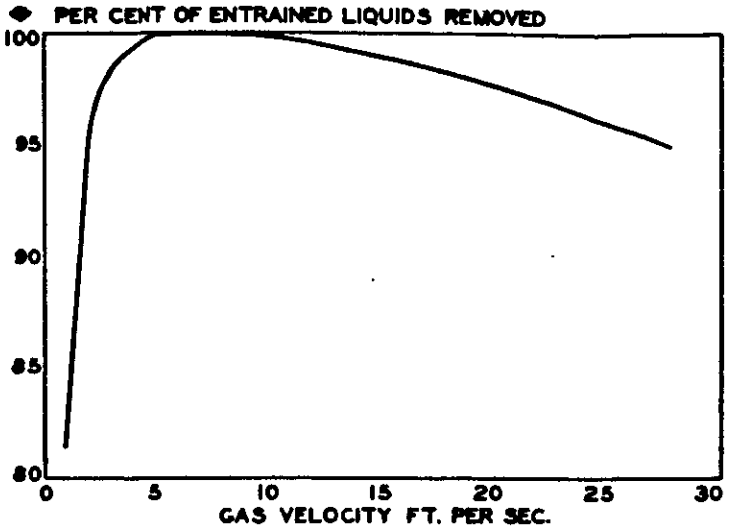
The element consists of wire knitted into a pad having a number of unaligned asymmetrical openings. Although similar in appearance to filter media, its action is somewhat different. The latter are rather dense and have small openings. This knitted wire, on the other hand, has about 97 to 98 per cent free voids and collects the particles primarily by impingement.

The material is available in single wound units of varying thickness in diameters up to 36 in., or in laminated strips for insertion through manholes in large process vessels. A view of the former is shown in Fig. 1.

How it works . . . The principle of this



KNITTED-WIRE-MESH screen used for mist extraction in small vessels. Fig. 1.



EFFICIENCY of wire-mesh mist extractors as a function of gas velocity. Fig. 2.

separator is similar to that of the vane-type unit. The gas flowing through the pad is forced to change direction a number of times although centrifugal action is not so pronounced. Impingement is the primary mechanism.

A liquid particle striking the metal surface, which it does not "wet," flows downward where adjacent wires provide some capillary space. At these points liquid collects and continues to flow downward. Surface tension tends to hold these drops on the lower face of the pad until they are large enough for the downward force of gravity to exceed that of the upward gas velocity and surface tension.

Velocity determines efficiency . . . Fig. 2 illustrates data obtained by Carpenter[1] on an evaporator operating with a salt solution. Operating experience tends to confirm the shape of the curve.

The impingement efficiency falls off rapidly at low velocities for the droplets will tend to drift between the wire. At high velocities the elements tends to flood. Liquid cannot flow downward against the increased upward velocity forces and therefore accumulates. As the interstices become full of liquid a portion is reentrained and passes out with the overhead gas.

Throughout the proper operating range the following equation accurately represents the data shown in Fig. 2:

$$V = K\,[(\rho_L - \rho_G)/\rho_G]^{1/2} \qquad (1)$$

where:

V = allowable velocity at separator pressure and temperature, ft. per sec.

ρ_L = density of liquid droplets, lb. per cu. ft.

ρ_G = density of gas, lb. per cu. ft.

K = proportionality constant.

For the average unit, values of K vary from 0.12 to 0.43, depending on the service and the location of the unit.[6] The value of K is a function of the disengaging space available between the source of the entrainment and the bottom of the pad. For oil and gas separators a value for K of 0.35 has proved satisfactory.

Best designs . . . Equation 1 is similar in form to those proposed for determining the allowable velocity in separators, absorbers, fractionators, etc. In these cases though, the value of K normally varies from 0.12 to 0.17. Consequently, the cross-sectional area of the pad should sometimes be less than that of the vessel proper. This is easily accomplished by blanking off a portion of the supporting grid.

Langmuir and Blodgett[2] concluded from their studies on impingement that the efficiency is a function of the number of targets presented. This may be accomplished by increasing the pad thickness, changing wire diameter or the closeness of the weave.

The wire mesh normally used falls within the following range:

Wire diameter—0.003 to 0.011 in.
Void volume—92 to 99.4 per cent.
Density—3 to 33 lb. per cu. ft.
Surface area—50 to 600 sq. ft. per cu. ft.

The most commonly used wire has a void volume of 97 to 98 per cent, has a bulk density of approximately 12 lb. per cu. ft., a surface area of 110 to 125 sq. ft. per cu. ft., with a wire diameter of 0.011 in. A pad thickness of 4 to 6 in. is sufficient for most separator applications although thicknesses up to 3 ft. have been reported. In separator service, where the guarantee requiring that the entrainment shall not exceed 0.1 gal. per M.M.c.f., 4 to 6 in. will normally suffice.

Materials used . . . Any common metal may be used in these units, including carbon steel, stainless steel, aluminum, monel, etc. The pressure drop is a function of the entrainment load, the pad design, and gas velocity but will not exceed 1 in. of water in the average installation. Because of this small pressure the elements do not have to be "held down" and are normally only wired to the support grid to prevent shifting.

Experience has shown that the support grid should contain at least 90 per cent free area in order to eliminate any restrictions to liquid drainage. The pads are light in weight so that a light angle-iron support is adequate.

The cost of knitted wire mist extractors obviously varies with the type of pad, its thickness, and the materials used. When using 18-8 stainless, $9 to $10 per sq. ft. will cover about 75 per cent of the installations. Carbon steel units cost about 25 per cent as much.

When both liquid and solids are present a portion of the latter will obviously be scrubbed out. When only dry solids are present, the efficiency is substantially less although design modifications have been at least partially successful. At the present time though this type unit is considered primarily a liquid unit.

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ELEMENTS OF FIELD PROCESSING—12

Centrifugal Mist Extractors

. . . Principles and operation

by Dr. John M. Campbell

IN previous parts of this series, it has been noted that centrifugal force has been an integral part of separation processes. The standard oil and gas separator has a tangential inlet, for example, that utilizes centrifugal force to separate out the larger droplets.

The same principle is used in some mist-extractor elements except that higher velocities are needed in order to separate the smaller droplets. Equations 6 and 7, Part 9, show that the velocity needed for separation is a function of the particle diameter, particle and gas density, and the gas viscosity. Equation 7 was written as:

$$D_p = \left[\frac{9\mu s}{\pi V_o N (\rho_L - \rho_G)} \right]^{1/2}$$

where:

D_p = diameter of particle that would be collected, ft. or cm.
μ = gas viscosity, cp.
s = radial path of particle, ft. or cm.
V_o = average velocity of gas, cm. per sec. or ft. per sec.
N = effective number of turns made by gas stream in the cyclone, dimensionless.
ρ_L = density of particle, g. per cc. or lb. per cu. ft.
ρ_G = density of gas, g. per cc. or lb. per cu. ft.

Velocity important . . . With a given system it is obvious that the size of particle collected is inversely proportional to the square root of the velocity. Consequently, the success of a cyclonic mist extractor is dependent on the velocity attained. Furthermore, the velocity needed to separate a given size of particle must increase as the density of the particle becomes less. In addition to producing the necessary velocity, the mist extractor must provide an efficient means of collecting and removing the particles collected in order to prevent reentrainment.

Need direction change . . . One common type of equipment is often called a "steam separator" since it has been widely used to separate condensate and pipe scale in steam systems. It is normally a relatively small vessel which imparts a high velocity to the incoming gas and then makes the gas change direction radically to prevent reentrainment. In general the simple type shown will separate particles 40μ and larger very efficiently.

Another type uses the same principle but, in addition, forces the gas to pass through a labyrinth which introduces impingement effects and forces the gas to change direction a number of times. This is in reality a combination type and is relatively efficient. The general performance characteristics are the same as efficient mist extractors of other types. Some, however, are complex and relatively expensive.

Cyclone becoming popular . . . In the last several years the use of a multicyclone extractor has become more widespread. A schematic view of this is shown in Fig. 1. This may be mounted in the top of a regular vessel or in a small separate scrubber.

This unit, marketed under the trade name of Aerotec, imparts a centrifugal force to the gas entering at A. It spirals downward and at vortex B reverses direction. Clean gas emerges at C while dust and distillates thrown to the wall by centrifugal force are discharged at D.

By using a number of such tubes in parallel, a high velocity is imparted to the gas which, by the above equation, would indicate that small particles could be separated. For a given flow rate, therefore, the number and size of tubes used depend on the relative density of the particles and the gas. With water, for example, less centrifugal force; i.e., more tubes, are used than with distillate.

The equation would also indicate that the separation would become more difficult as the gas density increased. This probably accounts for the lower efficiency exhibited by all centrifugal devices at elevated pressures. However, all other devices have this same failing. This common fault points out that increasing pressures give a corresponding increase in the extraction problem.

Pressure drop limited . . . Fig. 2 shows the over-all collection efficiency shown by the manufacturer when processing pipeline dust of the size shown. Similar

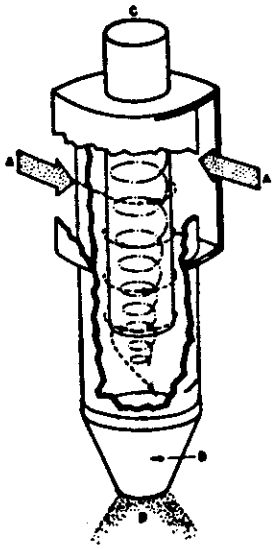


Fig. 1—Action of Aerotec tube.

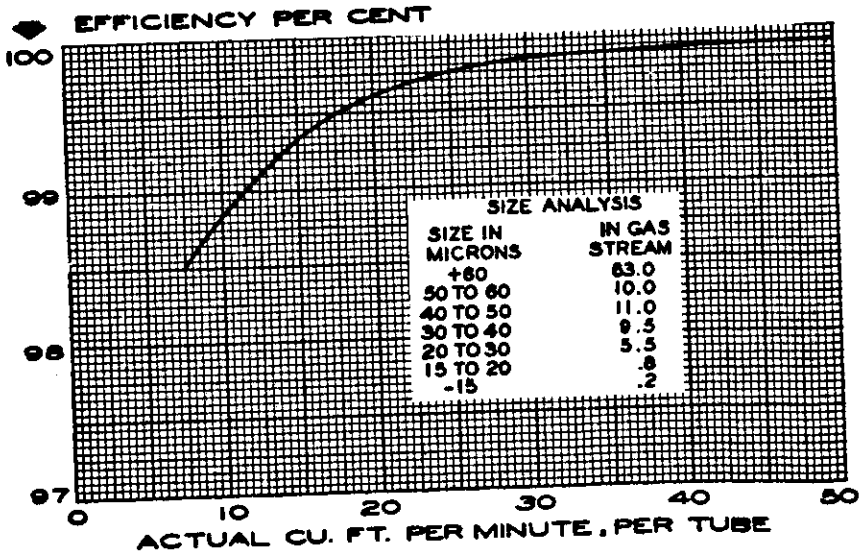


Fig. 2—Over-all collection efficiency vs. cubic feet per minute per tube for pipeline dust of size analysis as shown.

data show that the relationship between flow rate and pressure drop may be expressed by the following equation:

$$F = K (\Delta P)^{0.5} \quad (1)$$

where:

F = cu. ft. per minute per tube.
ΔP = pressure drop, psi.
K = function of pressure.

Typical values of K are:

Pressure (psig.)	K
20	22
50	14
100	10.5
500	5.4
700	4.2

This equation, together with Fig. 2, shows that the collection efficiency is a direct function of the pressure drop. The only practical limitation to the flow rate is therefore the allowable pressure drop.

Handles solids and liquids . . . The multicyclone separator has been used effectively in the separation of dusts as well as liquid particles. It is substantially self-cleaning and can handle relatively large quantities or both. In some cases an impingement-type element has been effectively used ahead of such a unit. The first acts as an agglomeration agent for the smaller particles which may then be separated in the centrifugal unit.

The chief disadvantage of any centrifugal unit is that the efficiency falls off rapidly as the velocity decreases below a certain point. Therefore, when widely varying loads are encountered this type of unit shows decreased advantage. However, if a higher pressure drop is tolerable the unit may be designed for the lowest flow rate anticipated since it gives efficient separation at greater than rated capacity.

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ELEMENTS OF FIELD PROCESSING—13

How Dust Filters Function

by Dr. John M. Campbell

NO discussion of mist and dust removal is complete without at least some mention of filtration. This is more often a pipeline, rather than a lease, application but the problem presented is at least partially of lease origin.

A filter differs from the mesh-type mist extractor in that the element is composed of rather densely packed woven elements that trap particles on the face of the pad as compared to the rather loosely woven metallic elements discussed in Part 10. The close clearance between the fibers allows the gas to pass while the larger particles build up on the surface and to some degree in the bed of the filter itself.

Materials . . . A number of materials have been used, including glass fiber, Dynel, felt, and similar materials that may be readily woven, or compacted, and formed into pads.

The primary use thus far has been for the elimination of dust in lines. One of the earliest applications was a "bag filter" downstream from dry-desiccant dehydration plants which, as the name implies, was primarily a porous woven bag held in place in the line. This rather crude attempt at dust separation was often more of nuisance than engineering value. As a cake of dust built up, the pressure drop became excessive and the bags often failed. Furthermore, the line had to be shut down periodically to permit cleanning of these filters.

Felt, compressed into pads, has also been used. The most successful applications had several layers of felt pads held in compression with the pads being placed parallel to the flow. All of these materials, however, were difficult to clean and tended to deteriorate under the action of the gas, particularly in the presence of liquids.

Part of this disadvantage has been overcome by the use of synthetic materials, such as glass fiber, which are more impervious to organic liquids.

Problem of liquids . . . Even with these materials, though, liquid separation appears low even though dust may be efficiently separated. Liquid particles appear to agglomerate on the downstream side of the pads as a solid film of liquid. As the gas passes through this film, bubbles are formed, burst, and are reentrained. At this point the problem is the same as the original one except that the dust is gone, and the liquid particles are larger and may be easier to catch in a conventional mist extractor. Observations such as this have led to the conclusion that the best system might well consist of a scrubber with both filter and mist extractor elements.

Desired properties . . . A filter element should ideally have the following characteristics:

1. Be self-cleaning to minimize down time.
2. Be readily removable for cleaning.
3. Be impervious to the action of organic liquids and water so that it does not swell or break down.
4. Have structural strength and be rigid enough to hold its shape.
5. Have a relatively low-pressure drop.
6. Have a surface that is "nonwetted" by the liquids encountered to prevent the liquid from "crawling" through the element.
7. Be so mounted that scale, drilling mud, and other large contaminants encountered are separated prior to the gas passing through the pad.
8. Be able to separate dust and liquid simultaneously with high efficiency.

Needless to say, the above animal is not in captivity. Many filters used have worked excellently in one location and given poor results in another. The results obtained with a given field installation are often difficult, if not impossible, to evaluate. After installation of a given filter a previous problem sometimes disappears, but one is never certain whether the filter or one or more other changes in the system is responsible. This undoubtedly accounts in part for the erratic results obtained.

Field type . . . Fig. 1 shows one type of commercial filter available on the market. It is composed of multiple circular elements, the number depending on the area needed. The gas flows into the top of the filter, through the elements, and out through the tubes. Dust is retained in the element and liquid coalesces into larger particles for removal in a mist extractor that may be located downstream. The end flange is removed for replacement of the elements. This type of filter offers the advantage of a high surface area per unit volume of vessel.

The amount of filter area needed is a function of the material, its density, the gas specific gravity and the method of construction. The following equation may be used to estimate the capacity of a given filter:

$$C = 38{,}400\ P^{0.47} \qquad (1)$$

where:

C = capacity in M.M.c.f. per day per sq. ft. of filter surface.

P = operating pressure, psia.

The capacity thus obtained may serve as a general guide for the sizing of filters if detailed information is not available for the specific element. If the element is located in the top of a separator, the capacity of the system may be limited by the vessel diameter as determined in Part 1

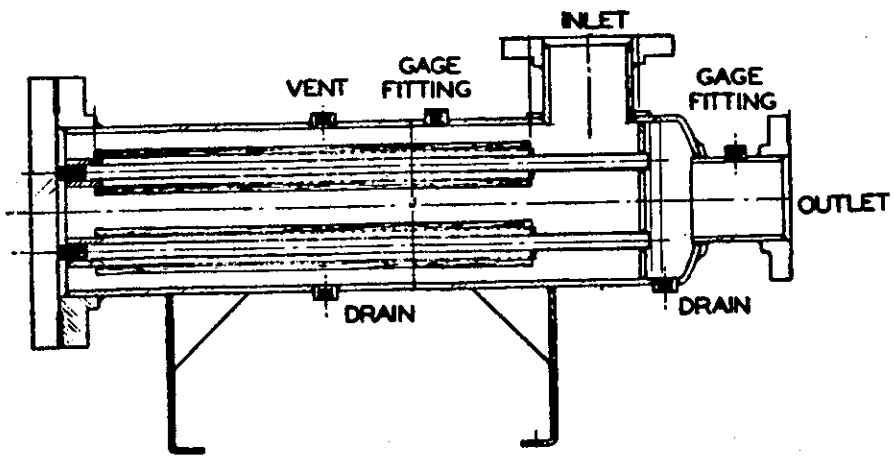


FIG. 1—Dust filter composed of multiple circular elements.

ELEMENTS OF FIELD PROCESSING—14

Oil-Field Instrumentation—Definition of Terms

by Paul S. Ache* and John M. Campbell†

A LARGE percentage of the routine problems encountered with lease equipment stems from failure or misapplication of controls. This has become even more pronounced in recent years as this equipment has become more complex.

Oil-field instrumentation is less complex than that in plants and refineries but operates under severe service conditions. In such service, the instruments often have wet supply gas only, infrequent adjustment by expert servicemen, and are exposed to the elements. Yet they must give continuous and trouble-free service. Despite these severe conditions, many troubles stem from improper application rather than inadequacy of the controls themselves, although many types are not suitable for oil-field use.

Control terms . . . In order to choose controls properly, one must know not only the requirements of the process but also the corresponding characteristics of the controls available. No understanding of controls may be complete without a familiarity with applicable nomenclature.

Pneumatic controls—those which are actuated by air or gas. (The most common type used in the field.)

Diaphragm motor valve—the term applied to a complete valve that uses pressure to open and close it.

Topworks (motor)—that portion of the motor valve which contains the mechanism to open and close the valve.

Innervalve assembly (trim)—includes the stem and seat within the valve body that actually controls the flow of fluid.

Fluid—any liquid or gas being controlled.

Proportional control—the system whereby the valve opening is proportional to the degree of change in the controlled variable.

Snap action—a mode of control whereby the valve is either wide open or fully closed.

Reset—an addition to a proportional control system to enable the instrument to hold itself at the control point as the process load varies.

Derivative response—a further addition that provides corrective action based on the time rate of change of the deviation from the control point.

Self-operated controller—a valve which is actuated directly by the controlled variable.

Pilot—a relay that transforms the controlled variable into an equivalent signal to the control valve, such signal controlling the action of that valve.

Supply gas—that gas necessary in a pneumatice pilot to operate it.

Controlled variable — the pressure, liquid level, temperature, or flow rate being controlled.

Measuring means—the means used to detect any change in the controlled variable.

Sensitivity—the ability to detect small deviations in the controlled variable.

Reproducibility—the ability of an instrument to repeat and measure consistently the value of the static condition over a period of time.

Static error—the difference between the absolute value of the controlled variable and the measured value.

Lag—the period of time by which the measured value follows the change in the absolute value of the controlled variable.

Static conditions—where all changes in the controlled variable are instantaneous.

Dynamic conditions—where the controlled variable is continually changing.

Normally closed valve—a valve that is held closed by a spring or some similar device and is opened by the action of pilot or the controlled variable or both.

Normally open valve—reverse of the above. One that is closed by the action of the pilot or the controlled variable or both.

Drift—this item includes reproducibility and the inability to repeat a measurement due to changes in the measured variable.

Process . . . Since the purpose of an automatic controller is to regulate a process, it is fundamental that the process be defined. A process is the collective functions performed in and by equipment in which a variable is controlled. As an example, a field heater which heats well effluent by hot circulating water in a unit of equipment in which the process of heating the well stream is accomplished. In the process is a controllable variable (the temperature) and a controllable medium (the well stream). Other controllable variables could be rate of flow, liquid level, or pressure. A control agent (circulating water) is the medium for effecting the temperature change, and thus regulation of the control agent regulates the controlled variable.

The total requirements of the process for the control agent at any one time is defined as the process load. If the rate of flow of the well stream increases, then more or hotter water is required to maintain the same temperature, and a change in process load has taken place. By the same token, a drop in inlet temperature at the original rate of flow would also constitute a process load change.

. . . Characteristics of a process include its potential, capacitance, resistance, and dead time. Capacitance, not to be confused with capacity, is the change in the quantity of energy or material per unit change in some variable, usually the controlled variable. The table below summarizes the characteristics of a process.

Characteristic—	Thermal	Pressure	Liquid level	Electrical
Capacity	B.t.u.	Cu ft.	Cu. ft.	Coulomb
Potential	Deg.	Psi.	Ft.	Volt
Capacitance	B.t.u./deg.	Cu.ft./psi.	Cu. ft./ft.	Farad
Resistance	Deg./B.t.u./sec.	Psi./cu. ft./sec.	Ft./cu. ft./sec.	Ohm

Dead time is that time lag which occurs when energy is being transferred at a constant rate of flow through a given distance. It is equal to the time it takes the energy to move the distance. The dead time is a characteristic of the process, and it is not to be confused with the time lag inherent in the automatic controller itself.

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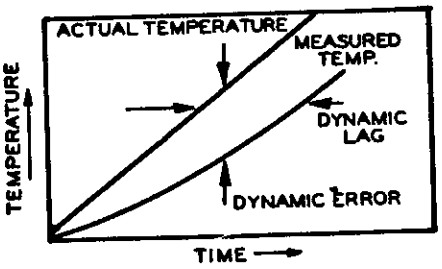
ELEMENTS OF FIELD PROCESSING—15

Oil-Field Instrumentation—How Controllers Work

by Paul S. Ache* and John M. Campbell†

PURPOSE of the automatic controller is to prevent deviation of a process from a desired standard. This may be accomplished by various modes of control action, each of which incorporates a distinct limit range, and speed of correction.

Choice of modes depends on the frequency and magnitude of the process load changes, the degree of control needed and the dynamic lag inherent in the process and controls used.



Proportional control . . . This is the basic action that is employed in all controllers not using snap (on-off) action. It is defined as control action in which a continuous linear relation exists between the value of the controlled variable and the position of the valve. The ratio of change of pilot pressure or electric current to a unit change of the controlled variable is the sensitivity:

$$dP/dt = S\,(dy/dt) \qquad (1)$$

where:

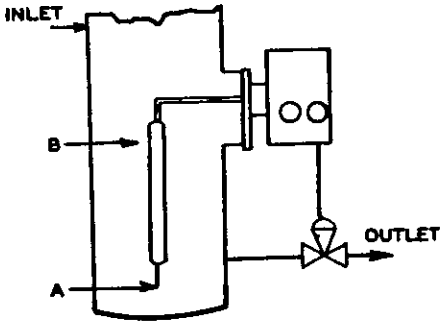
dy/dt = rate of change of the controlled variable with time (level, pressure, etc.)

dP/dt = rate of change of pilot output pressure or current signal with time

S = characteristic sensitivity constant of the pilot relating output to input

The proportional band or range, expressed as a percentage, relates the percentage of the full range of the measuring means that the controlled variable has to traverse to fully stroke the valve. This may be illustrated by the figure at top of next column.

This represents a typical level-control application utilizing a displacement-type controller, i.e., one which does not float on top of the liquid but depends on the varying buoyancy of the liquid as the level changes. Consequently, the control range is the distance between points A and B. When the level is below A or above B, a change in level has no effect on the buoyancy of the float.



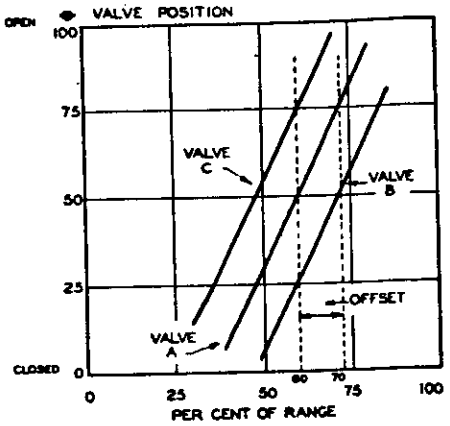
Therefore, at 100 per cent proportional control the level would have to move from A to B, or vice versa, to fully stroke the valve from full open to closed. At 25 per cent, a level change of 25 per cent of the distance A to B would fully stroke the valve. The same principles apply with all other modes of control.

Example: A back-pressure controller with a range of 0 to 100 psig. is set at 50 per cent proportional control, using a normally open valve. If the valve is open at 100 psig., (a) at what pressure will it be fully closed, and (b) what is the opening at 75 psig.?

Solution: The control range is 100 psig. which at 50 per cent proportional control means that a (0.50) (100) = 50 psig. change will fully stroke the valve; therefore it is fully closed at 50 psig. At 75 psig. it would be half open.

Control of flow . . . The amount of fluid that a valve will pass at a given position depends on the size and characteristics of the inner-valve assembly. Ideally the amount of fluid passed under given conditions, with a given valve, should be directly proportional to the valve opening. Often it is not, however, so that the type of inner valve has a definite effect on the proper proportional setting. Also ideally, the diameter of the inner valve should be such that the range of anticipated process loads may be handled when the valve is between 25 and 75 per cent open. In the final analysis, the choice of the proportional band for a given installation must be governed by experience and adjustment on the job.

This is illustrated in Fig. 1. The slope of the lines depends on the proportional band used, each line representing a valve of given size. Under these conditions each valve may operate only along a single line, in accordance with Equation 1.



SIZE AND CHARACTERISTICS of inner-valve assembly determine amount of fluid a valve will pass at a given position. Fig. 1.

If the instrument were set to control at 60 per cent of the range, valve A would be 50 per cent open, valve B would be 25 per cent open, and valve C would be 75 per cent open.

Suppose, however, that an instrument set to operate along line A is in liquid-level service and the flow rate through the vessel increases such that a valve opening of 75 per cent is needed to maintain the level. According to line A the only way the proportional controller can provide this opening is for the level to be at 70 per cent, or 10 per cent above the set point. This 10 per cent is called offset.

Consequently, with proportional control alone a change in process load brings a change in the valve position and some change in the liquid level. This proper proportional setting is therefore one which makes the lines in Fig. 1 as near perpendicular as possible.

Zero per cent proportional control represents a special case. It is the upper limit of a range where the liquid level is controlled by the valve alternately opening and closing. This is normally called snap action or on-off control. At a given snap setting the valve will stay closed until the level reaches a certain point between A and B and then remain full open until the level drops to the control point. Where rapid process load changes are encountered, such as surging flow through separators, snap action is normally recommended.

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FIELD PROCESSING

More on

How Field Controllers Work

Reset action enables controller to handle load change without changing control point

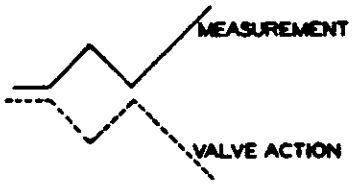
by Paul S. Ache* and John M. Campbell†

THE "offset" obtained with proportional control is sometimes too great to be tolerated and it is necessary to add other modes of control. One such addition is known as reset.

Figs. 1 and 2 show on-off action



ON-OFF ACTION of a controller. Fig. 1.



PROPORTIONAL ACTION of a controller. Fig. 2.

and proportional action respectively. In the latter it is seen that the amount of valve movement is proportional to the deviation of the measured variable. By contrast the amount of correction applied with reset action depends both on the magnitude and duration of the deviation away from the control point. The prime purpose of reset is, therefore, to prevent offset and keep the controlled variable at the control point even as the process load changes.

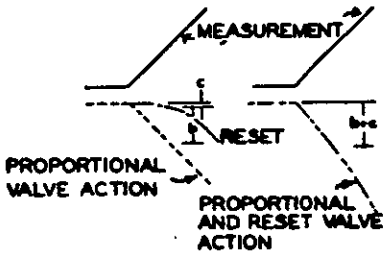
Adding reset action . . . The differential equation of a controller with proportional plus reset action is:

$$dP/dt = S\,(dy/dt) + SR\,(y - y_o) \quad (1)$$

where dP/dt, S, and dy/dt are defined in Part 15, R is controller reset constant, and $y - y_o$ is deviation of the controlled variable from the control point.

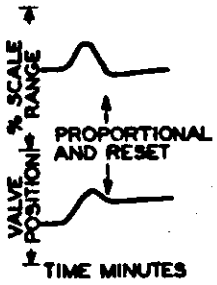
In this type of controller the two corrections occur simultaneously as shown in Fig. 4. Fig. 3 shows the action if they took place separately.

Fig. 5 shows typical curves in a process being controlled by this combination action. When a change in process load takes place the valve returns the measurement to the control point with a minimum of cycling. The original motion of the valve corresponding to the measurement change is due to proportional action, but the change of the valve to its new position is due entirely to reset. In other words, the valve has moved to a new position to maintain the controlled condition. With only proportional control both the valve and controller would have changed.



VALVE ACTION for reset and proportional controllers (at left). Fig. 3.

VALVE ACTION for combined proportional-reset controller (at right). Fig. 4.

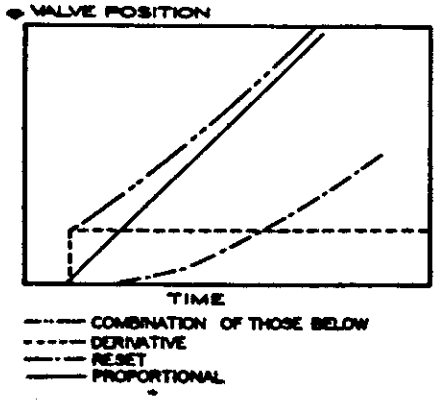


TYPICAL CONTROL CURVES when using a proportional-reset controller. Fig. 5.

Derivative response . . . Proportional control plus reset does not provide correction that is rapid enough for certain processes. Derivative response may therefore be added to anticipate a change in process load and tranmit a corrective signal to minimize the lag. This action corrects on the basis of the rate of change of the deviation from the desired standard. This term stems from the fact that the first derivative of the change from the desired standard is incorporated into the control mechanism. The equation for an instrument with proportional plus reset plus derivative response is therefore:

$$dP/dt = SR\,(y - y_o) + S\,(dy/dt) + ST\,(d^2y/dt^2) \quad (2)$$

where T is an instrument adjustment factor. The action of a controller with this response is shown in Fig. 6.



TYPICAL CONTROL CURVES for proportional, reset, and derivative-action controllers. Fig. 6.

The general applications of the various combinations of control action may be summarized as follows:

Proportional—where process time lag is small in comparison to the apparatus capacity such as tank heating, large surve vessels, etc., or where "offset" may be tolerated.

Proportional plus reset—where it is necessary to use a narrow band to prevent "hunting" or overcontrol, and as the frequency and magnitude of the process load changes become greater.

Proportional plus derivative response — in processes involving long-time lags and large capacities when small and frequent load changes occur. Does not provide compensation for process load changes.

Proportional plus reset plus derivative response—where long-time lags and large capacities are combined with large and sudden load changes.

Inasmuch as the addition of reset and derivative response increases the investment cost, they should not be utilized unless their action is needed for proper control.

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ELEMENTS OF FIELD PROCESSING—17

How to Select Liquid-Level-Control Instruments

Best bet is to choose simplest control that will meet process requirements

by Dr. John M. Campbell

CONTROL of liquid level is an integral part of field processing. In view of the widespread applications and the variety of conditions encountered, it is not surprising that a number of mechanisms are used. These may be conveniently subdivided as:

1. Mechanically operated valve actuated by a float.

2. Pilot-operated valve actuated by a float.

3. Diaphragm motor valve actuated by:
(a) Displacement type controller.
(b) "Floatless" level controller.

4. External devices, including inverted bucket traps, float traps, etc.

Either snap action or proportional control is normally employed on most lease equipment. Most applications do not justify the cost of reset and/or derivative response since small changes in level with flow rate, and time lag, are usually not critical problems.

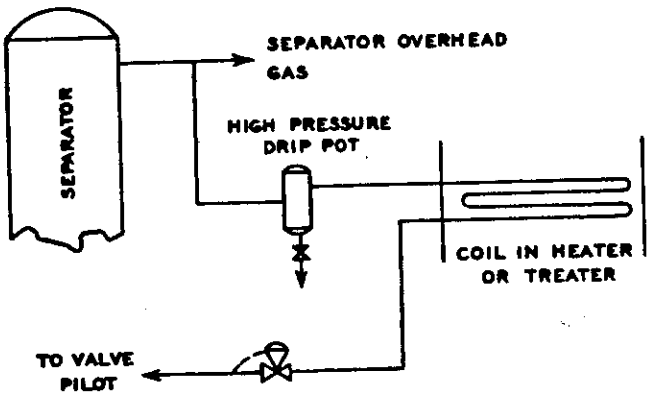
Separator controls . . . These are usually set on snap action because this enables the controller better to handle a surging condition. The degree of snap action will depend on the conditions encountered. However, it should be set so that not over 25 per cent of the vessel's liquid capacity is filled, above the control point, before the valve opens.

The inner-valve size should be determined on the basis of 110 per cent of the maximum flow to be encountered. This means that during a surging condition the valve can handle the volume and prevent the separator from filling up.

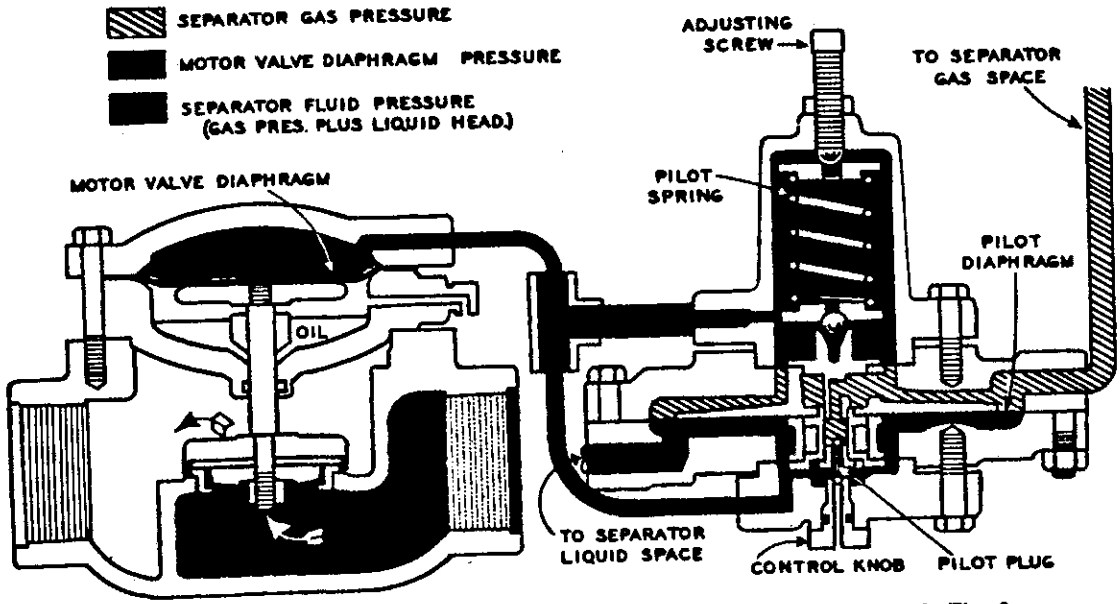
Some uncertainty always exists, of course, when one tries to estimate future flow rates. On low-pressure separators, sizing is not a serious problem because of the low pressure drop across the valve. On high-pressure gas-condensate separators, however, where the liquid flow rate is low, too large an inner valve on snap action can blow all the liquid out of the separator and allow gas to enter the stock tank before the valve can close.

The author observed one such installation when 1,000-psig. gas hit a 100-bbl. tank, knocked the cover off the vent valve, sprayed oil over the countryside, and lifted the tank off the ground.

Sizing curves . . . The manufacturers of control valves furnish sizing curves and



TYPICAL INSTRUMENT INSTALLATION for separator-overhead gas. Fig. 1.



PILOT IS ACTUATED by the varying liquid head in the floatless level control. Fig. 2.

"Ability to make repairs with field personnel will reduce down time"

these should be consulted. In using these it should be remembered that critical flow conditions occur when the upstream pressure is approximately twice the valves' downstream pressure. Consequently, only twice the downstream absolute pressure should be used with these sizing curves if the actual pressure ratio exceeds two.

When handling very small volumes of liquid, the use of such curves often indicates the use of so-called "metering trim." Its use is not recommended in field applications for the small quantities of solid materials often present may clog the valve. For this reason nothing smaller than ⅛-in. needle trim is usually recommended in field service.

Float-operated mechanical oil valves are satisfactory at pressures up to 125 psig. Such valves are directly actuated by movement of the float on top of the liquid, through an adjustable linkage. At higher pressure the leverage supplied by the float is insufficient to provide satisfactory valve action because of the pressure acting on the valve seat.

On separators with greater than 125 psig. working pressure, the use of a pilot-operated valve is advisable. Whenever applicable this type of control is very dependable and simple to adjust and repair.

Float-actuated pilot-operated valves are particularly applicable at pressure to 1,000 psig. on vertical vessels. Such control has been "standard" on separators for years because the pilot is rugged, simple, and will operate satisfactorily with the "wet" supply gas from the separator overhead.

This overhead gas contains entrained liquid so that a drip pot ahead of the pilot is advisable. With high-pressure separators the expansion of gas from separator to instrument pressure may cause sufficient temperature drop to condense some water. This in turn presents hydrate or freezing problems, particularly in cold weather.

A variety of solutions have been used, including tracing with warm separator gas, use of dehydrator pots, running such gas through a heater or treater if available, and insulation of the lines. A typical instrument gas installation is shown in Fig. 1.

Displacement-type liquid-level controls utilize the buoyant effect of liquid on the float. The average float movement does not exceed ⅛ in. As the level varies on the float, the weight changes correspondingly and this change in torque is transmitted to the pilot which, in turn, controls the valve movement.

This type of control is applicable in all pressure ranges but is used primarily at high temperatures or in horizontal vessels. Because of the small movement and small float diameter, the vessel opening may therefore be decreased. This advantage becomes pronounced at high pressure or on small-diameter horizontal vessels.

The pilots on such controls are more sensitive than those discussed above which allows one to control the level closer. Most such pilots also allow the valve action to be changed from snap to varying degrees of proportional control by a simple adjustment.

They are necessarily more complex and expensive. Consequently, their use must be justified by process requirements. The instrument gas requirements are also more critical, for as little as one drop of water may plug the small orifice in some pilots.

Floatless level controls are relatively new. With these the pilot is actuated by the varying liquid head in the vessel, as shown in Fig. 2. As the liquid rises in the separator it overcomes the pilot spring and forces the pilot assembly upward, closing the upper seat and opening the lower separator seat which vents the diaphragm pressure to atmosphere. The separator fluid pressure then opens the valve. When the valve is throttling, the nonbleed three-way valve action of the pilot plug against its seat adjusts the motor valve diaphragm pressure.

The system shown is applicable up to 125 psig. although similar models are available to 4,000 psig. This type of control has the obvious advantage of eliminating large vessel openings and offering application on very small vessels.

External devices such as float cages, traps, etc., find application particularly on vessels with small or infrequent liquid loads. Some types of float cages offer no particular economic advantage at high pressure. They are most commonly used on low-pressure plant suction and instrument systems. Inverted bucket traps have been successfully used on small glycol absorbers for economic reasons but regular level controls certainly offer advantage.

At high pressures a choke nipple downstream from the control valve is advisable, for it:

1. Provides a factor of safety if the valve "cuts out" or fails to close for any reason.
2. Reduces the pressure differential across the valve, which improves the valve action and enables it to shut off tighter.
3. Prevents damage to low-pressure equipment downstream if the instrument supply gas fails on a normally open valve.

The control range with a float-operated controller is limited by the flange diameter and the float arm length. With a displacement element control is only possible throughout the float length, for a change in level above or below the float does not affect the buoyancy. Therefore, the desired change in level should not exceed the float length.

The choice of a liquid-level control is somewhat arbitrary but it is good policy to choose the simplest control that will meet process requirements. Average lease-operating conditions impose severe service on the control and the ability to make repairs with field personnel will generally reduce down time.

ELEMENTS OF FIELD PROCESSING—18

Back-Pressure and Pressure-Reduction Control—1

by Dr. John M. Campbell

THE regulation of back pressure and pressure reduction in a system may be conveniently divided into three categories when considering the type system needed. The low-pressure range is usually 0-125 psig., the intermediate-pressure 125-500 psig., and the high-pressure greater than 500 psig. The use of these ranges is primarily for convenience since some types of valves can operate satisfactorily in all of them.

All pressure regulators are similar in principle, the specification of type being dependent on the process requirements, pressure drop, variation in flow rate, limitations of the loading device, and the maximum pressure. In either of these services pressure is regulated by the control of flow rate.

This flow is controlled through movement of the regulator inner valve which is held either open or closed by some means of preloading. The amount of preloading and the size of the diaphragm used, if any, are such that the inner valve will move to the opposite extreme of travel shortly after the diaphragm pressure passes the desired working pressure. The control pressure is therefore varied by changing the amount of preload, thus upsetting the equilibrium between it and the diaphragm pressure. With most valves increasing preload increases pressure.

The preloading may be accomplished through the use of spring compression, dead weight or fluid pressure. Valves A and B, shown in Fig. 1, are examples of back-pressure valves using the first two forms of preloading. Valve C is a spring-loaded valve incorporating a built-in pilot. All three valves shown are suitable in the low-pressure range.

Back-pressure service . . . In back-pressure service the preloading normally closes the valve, while in pressure reduction the valve is normally open. With the former the upstream pressure is introduced under the diaphragm and works against the loading device to position the valve properly.

In pressure-reduction service, the downstream pressure is introduced under the diaphragm and tends to close the valve until the inner valve is properly positioned. Many valves may be changed from one service to the other by simply reversing the valve body and the stem, while still others have provision for reversing the diaphragm action.

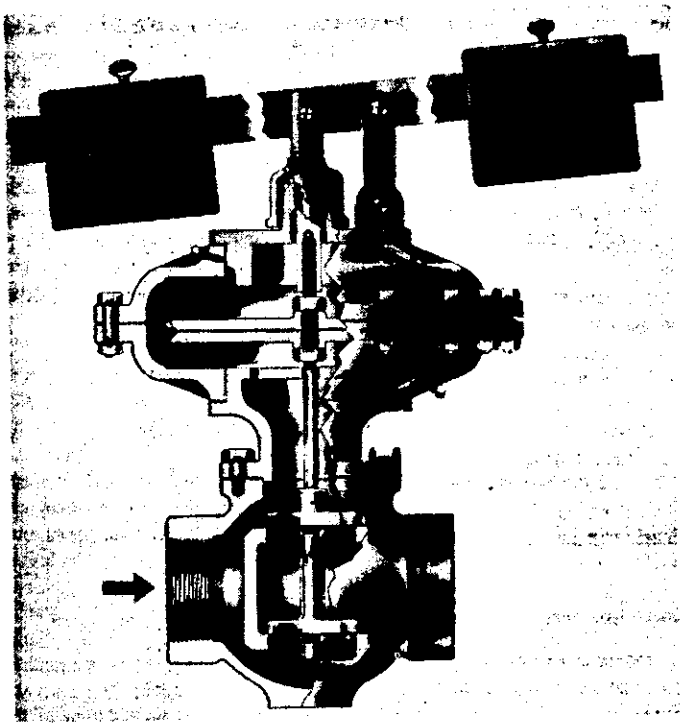
Some applications require that a certain differential pressure be maintained across the valve regardless of how much the upstream or downstream pressures vary. The chief application of this is with positive-displacement pumps where the maximum pressure ratio is controlled by a valve located on a bypass.

In this service the valve is normally closed with the downstream (higher) pressure under the diaphragm and the upstream (lower) pressure on top of the diaphragm. The desired differential is then established with the loading device.

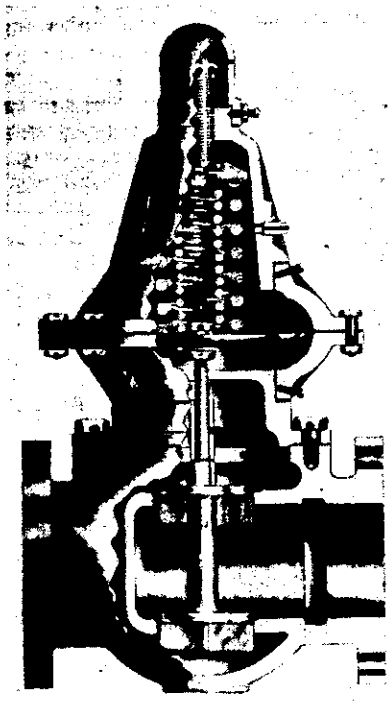
Each of the types of preloading has its distinct applications and limitations. The weight-loaded regulator is very sensitive within its pressure range if the flow rate does not vary widely. This is primarily true because the inner valve may move through a greater change of position than with a spring-loaded valve. The greater sensitivity is particularly pronounced when the diaphragm is in direct communication with the valve body, which eliminates friction in the stuffing box.

Very few weight-loaded regulators are recommended above 50 psig. because at greater pressures heavy weights are necessary. As a result a small change in controlled pressure is only a small portion of the loading weight and there is less subsequent movement of the inner valve, which in turn renders control more difficult.

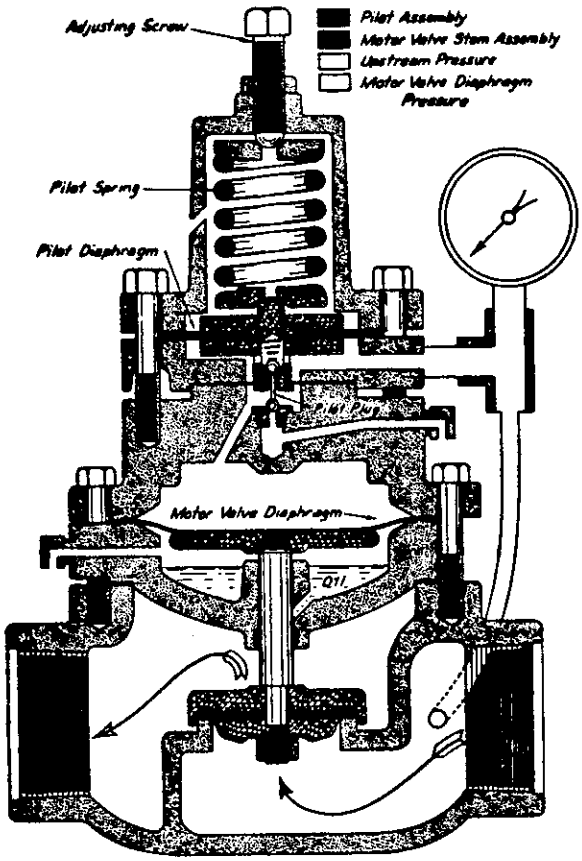
Sensitivity . . . A spring-loaded regulator is generally slightly less sensitive than a comparable weight-loaded regulator under constant-flow conditions. In low-pressure service (particularly between 0 - 20 psig.) it is an ideal type of loading for the springs used tend to compress in direct proportion to the load, with constant load increase being required for each increment of



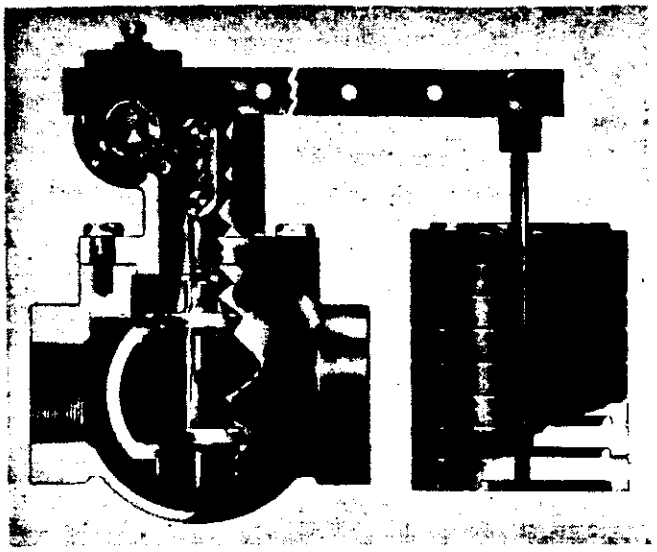
DEAD-WEIGHT PRELOADED DIAPHRAGM back-pressure valve. Valve A. Fig. 1A.



SPRING-COMPRESSION valve. Valve B. Fig. 1B.



SPRING-LOADED VALVE, incorporating built-in pilot. Valve C. Fig. 1C.



SIMPLE, WEIGHT-LOADED back-pressure valve. Fig. 2.

compression. Consequently, spring loading is excellent in low-pressure systems and where the diaphragm is loaded with a pilot. As each compression increment is a valve-travel increment, the valve travel is always in proportion to the relationship of the controlled pressure to the preloading.

At high pressure this characteristic does not exist because of the heavy springs required. Therefore nonpilot-operated spring-loaded regulators are not normally recommended above 125 psig. because of this loss in sensitivity and their closer approach to the price of pilot-loaded regulators.

With either a direct-acting weight or spring-loaded regulator a change in the uncontrolled pressure will cause a corresponding change in the controlled pressure if the pressure drop is low. This is true especially if the pressure drop is less than 25 per cent of the inlet pressure. Therefore when accurate control is necessary under these circumstances, pressure-loaded or pilot-operated valves are recommended.

Throttling and shutoff . . . Both the regulation of flow and tight shutoff should not normally be attempted with the same valve unless absolutely necessary. If this is required by operating conditions, a single-seated valve should be used. Double-ported valves are seldom operated under the actual conditions at which they were manufactured and tested. Consequently, as temperature changes the distance between the two seats, only one of the two will seal off tight.

Single-ported valves will seal off tight but the unbalancing force on the inner valve increases with the square of diameter. Inasmuch as the diaphragm-spring combination must overcome this force and still control travel, it is logical that the inner-valve size is limited by practical diaphragm size. As a general rule of thumb, direct-acting spring and weight-loaded regulators up to 2-in. diameter may be used with single-port construction.

Pilot-loaded valves, either pressure-balanced or spring-loaded, offer particular advantage at higher pressure since the pilot presents a means of increasing the control-valve travel with a given change in controlled pressure. This enables one to use a more accurately sized valve. The resultant smaller valve gives better shutoff and closer control since the full range of valve travel my be utilized.

Low-Pressure Service

It is in the low-pressure range that weight and spring-loaded regulators find particular application on the lease. One of the most common applications is in back-pressure service on low-pressure separators operating at less than 40 psig.

TYPICAL LOW-PRESSURE-VALVE COSTS (2-in.)

Valve A, Fig. 1	$140
Valve B, Fig. 1	85
Valve C, Fig 1	90
Fig. 2	50

All of the valves shown in Fig. 1 are applicable as is that shown in Fig. 2. The latter is similar to a diaphragm-type weight-loaded valve except that the weight simply acts to counteract the unbalanced force acting under the inner valve. The choice of this valve is predicated primarily from cost considerations since it is the cheapest of the valves, as the comparison below shows.

The simple weight-loaded valve shown in Fig. 2 has the advantage of being cheap and simple. Its primary disadvantage stems not from its operation but the circumstances surrounding its use. The weights become loose and shift, are lost, or are hindered in movement by outside obstructions. It is not too uncommon to see rocks and other miscellaneous objects used as substitutes for, or additions to, the proper weights. Consequently, these regulators serve as a proper but not necessarily satisfactory back-pressure control.

Not too many spring-loaded regulators are used on such separators because of their higher price, although they are generally satisfactory below 40 psig. The best regulator in this range is Valve C, Fig. 1. It gives fine control throughout the range with widely fluctuating flow rates and uncontrolled pressures. Where fine control is necessary, particularly above 40 psig., the extra investment is normally justifiable.

Because of the almost infinite number of different field conditions that arise, the above considerations can at best only serve as a general guide. Unfortunately the actual choice of type must often stem from experience.

ELEMENTS OF FIELD PROCESSING—19

Back-Pressure and Pressure-Reduction Control—2

by Dr. John M. Campbell

AT pressures above 125 psig. it is usually difficult to justify anything other than a pilot-operated diaphragm motor valve for pressure control. Non-pilot loaded valves are sometimes applicable but usually only in those circumstances where pronounced load changes are not encountered. At these higher pressures the cost differential also becomes less, which further encourages the use of pilot-operated controls.

Most pilots now in use in the oil field are pneumatic in nature and use natural gas as the actuating fluid. Any natural gas that is free of fluids and at a pressure greater than 15 psig. is suitable in this service.

Pilot Workings

Fig. 1 is a schematic view of a pressure pilot that will give both proportional and on-off control. As the controlled pressure varies the bourdon tube (A) will change shape and in turn raise or lower the curved bar (B).

The supply pressure is held constant between 15 and 20 psig. by pressure regulator (F). The vent (C) is so sized that when wide open (curved bar away from it) it will pass more gas than orifice (E). Consequently the pressure on the valve diaphragm and the valve position depend on the opening of vent (C) which in turn depends on the position of the curved bar, as fixed by the bourdon tube.

If the vent is wide open the pressure on the diaphragm is zero while if it is fully closed the diaphragm pressure equals the controlled supply pressure.

Limited vent . . . It is necessary that both the vent and orifice be very small in order to minimize the amount of gas vented. If the curved bar (B) had a fixed pivot at (G) rather than a bellows, only on off (snap) action would be possible because of these small openings. From a purely mechanical standpoint any small movement of the curved bar would in effect make the vent wide open.

The bellows (D) is used to impart a rotating motion to the curved bar around the end of the vent. When the bar begins to rise off the vent the bellows contracts which tends to keep part of the vent opening covered. As a result more vertical movement of the bar is necessary to fully open (or close) the valve. All in-between points then represent some degree of proportional control.

The vertical distance that the bourdon tube has to lift the bar to completely stroke the valve increases as the lever arm represented by the distance between (C) and (D) decreases. This then allows for adjustment of the proportional band.

If the vent is close to the left end of the curved bar the movement of the bellows imparts little rotation to the bar and on-off action results.

Control markings . . . Most pressure pilots have the bar marked to show various percentages of proportional control. The percentage shown indicates that the controlled variable must vary through that percentage of the instrument's range to fully open or close the valve.

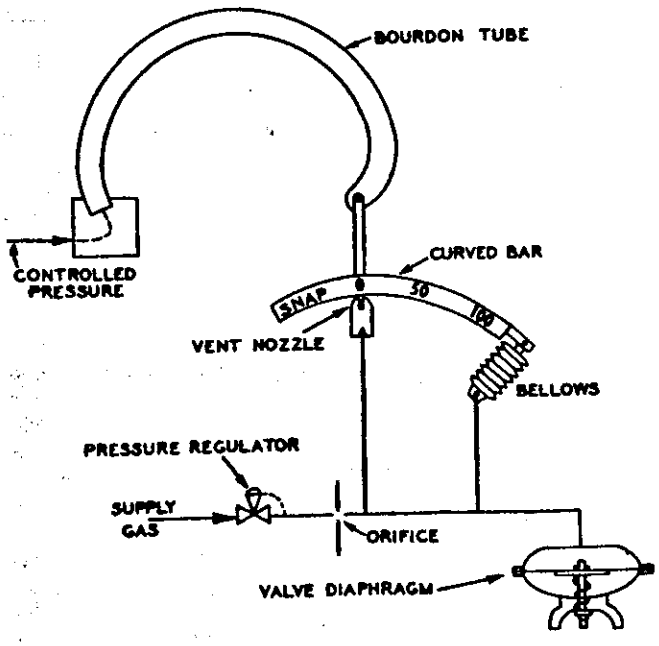
Example—If a pressure pilot has a bourdon tube with a range of 0-200 psig. and it is set on 50 per cent proportional control how much must the pressure vary to make the valve be fully stroked?

Answer—It must vary (0.50) (200 — 0) = 100 psig. to fully stroke the valve.

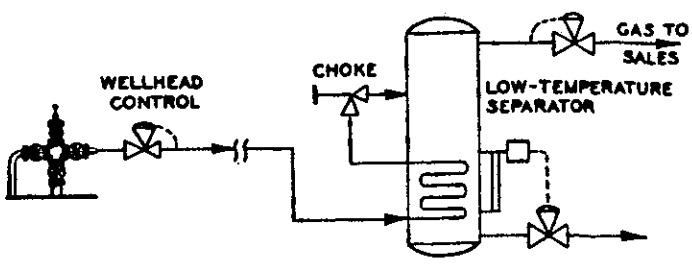
It therefore follows that the closer the vent is to the bellows the higher the precentage of proportional control obtained.

Most pressure pilots are slightly more complex than the schematic presentation in Fig. 1, in order to improve the mechanical action and the sensitivity. However, the basic mechanism and principle of operation shown are common to all pneumatic pilots.

Balanced-diaphragm pilots . . . In most applications the valve is spring-loaded but there are some where a large quantity of high-pressure gas must be controlled with a low-pressure drop. Such an application is encountered with the switching valves of a dry desiccant dehydrator. A large inner-valve assembly is required which increases the unbalanced forces on the diaphragm,



PROPORTIONAL PILOT operated by gas allows variable valve positioning depending on controlled pressure. Fig. 1.



PRESSURE CONTROL for high pressure wells keeps constant separation pressure. Fig. 2.

which in turn necessitates a heavy spring.

In such cases the use of a pressure-loaded balanced diaphragm valve is indicated. With this type greater valve travel may be obtained, with less change of controlled pressure, than with any other type. With a constant loading pressure on a larger balanced diaphragm the available force to travel the valve with incremental change of control pressure is greater than a spring-loaded valve due to spring characteristics. However, spring loading is simpler wherever applicable.

The greater the pressure drop across a valve the greater the unbalanced force that exists. Where this pressure drop is greater than several hundred pounds per square inch it is good practice to install a choke nipple downstream from the valve. This choke nipple should preferably be about 10-25 per cent larger than the inner valve so that the latter is the controlling element.

The nipple serves two purposes: (1) It reduces the pressure drop across the valve and increases sensitivity, and (2) it acts as a secondary control in the event the valve fails to close or cuts out.

The pilot shown in Fig. 1 may be modified to give what is known as high-low shutoff. A pilot so modified will cause the valve to close if the controlled pressure becomes higher (or lower) than the preset condition. This modification has particular application in high-pressure wells to:

1. Shut in the well in the event a line break is encountered.
2. Prevent overpressuring of a separator or other piece of equipment downstream from a choke or valve.

Automatic choke . . . In some instances it is desirable to use an automatic rather than a manual adjustable positive choke. A regular diaphragm motor-valve top-works may then be combined with the choke body to give a control valve and choke combined. Although it costs more there is some advantage to using the positive choke as a rough control with a separate valve at the wellhead for safety.

Fig. 2 illustrates such a scheme where several wells are being flowed to a central low-temperature separation unit, the pressure drop being taken at the unit. The valve at the well protects against line break while the back-pressure valve at the separator holds the separation pressure constant regardless of how the sales line pressure varies. The two control valves fix the pressure drop across the choke with the flow rate being determined by the choke setting.

Where very high formation pressures are encountered one or more "storm chokes" in the well proper makes a desirable installation.

ELEMENTS OF FIELD PROCESSING—20

What you ought to know about

Separation of Oil and Water

by Dr. John M. Campbell

INASMUCH as oil and water occur together in the reservoir their separation at the surface is a basic problem. They are substantially insoluble so that gravity separation is possible even when they are emulsified and chemical and/or heat treatment is necessary.

Equipment for performing this separation must have provisions for controlling the level of the interface between the oil and water, the top of the oil level, and the vessel pressure. The interface level may be controlled by the standard liquid-level pilot equipped for such purpose.

However, the control is less positive because the densities of the two liquids are closer than liquid gas in the normal separation. However, the chief drawback to that type of control is cost. Consequently other methods have been developed.

Simple system . . . The simplest separation system is that represented by a low-pressure free-water knockout of the type shown in Fig. 1. The free water settles out to the bottom and the oil and gas pass overhead—usually to a treater. The outlet B is extended into the vessel to insure the presence at all times of a gas cap and to control the oil level.

The location of the oil-water interface E is fixed by the position of adjustable nozzle D. Since the water outlet box is vented into the gas cap, rotation of this nozzle changes the interface level for:

$$d_o\rho_o + d_w\rho_w = S_w\rho_w + \Delta P_f \quad (1)$$

Where:

d_o = thickness of oil in vessel, ft.
d_w = height of water in vessel proper, ft.
S_w = height of water in outlet box.
ρ_o, ρ_w = density of oil and water respectively, lb. per ft.^{3}
ΔP_f = pressure drop in downcomer pipe and outlet box, lb. per ft.^{2}

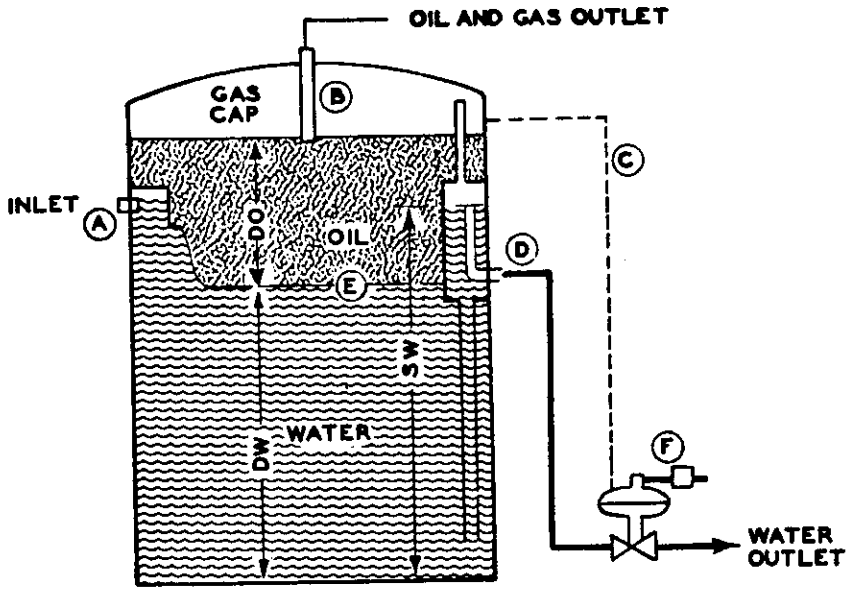
In a well-designed system operating at normal capacity ΔP_f may be neglected. Equation 1 may then be rearranged to read:

$$d_w = S_w - d_o\,(\rho_o/\rho_w) \quad (2)$$

Consequently any change in S_w will cause a corresponding change in d_w and d_o. As S_w is decreased d_o gets larger so that small changes in S_w give correspondingly larger changes in d_w.

Water removal . . . Water is removed from the vessel with a simple weight-loaded, self-acting diaphragm valve. With the top of the diaphragm vented into the vessel gas cap the only force acting on the underside of the diaphragm is the head of liquid. The weight is then simply adjusted on the arm to compensate for the desired liquid head operating the valve.

This is really a form of back-pressure service but only a valve designed specifically for such service should be used. No back-pressure valve is shown



SIMPLEST SEPARATION system is this low-pressure free-water knockout where water settles to the bottom and oil and gas pass overhead. Fig. 1.

on the oil-gas line because this type of vessel often floats on the line between the well and treater. If this were not the case such a gas back-pressure valve would be needed.

Fig. 2 shows a similar system for a treater. In this case though an additional valve is necessary to remove the oil which is separated from the gas in the top section. This valve operates in the same manner as the water valve.

The water syphon G is a type commonly used and the principle is the same as the rotating nozzle. It must also be vented back into the gas space. The oil outlet is a fixed opening so located that a gas cap may be preserved below the top plate.

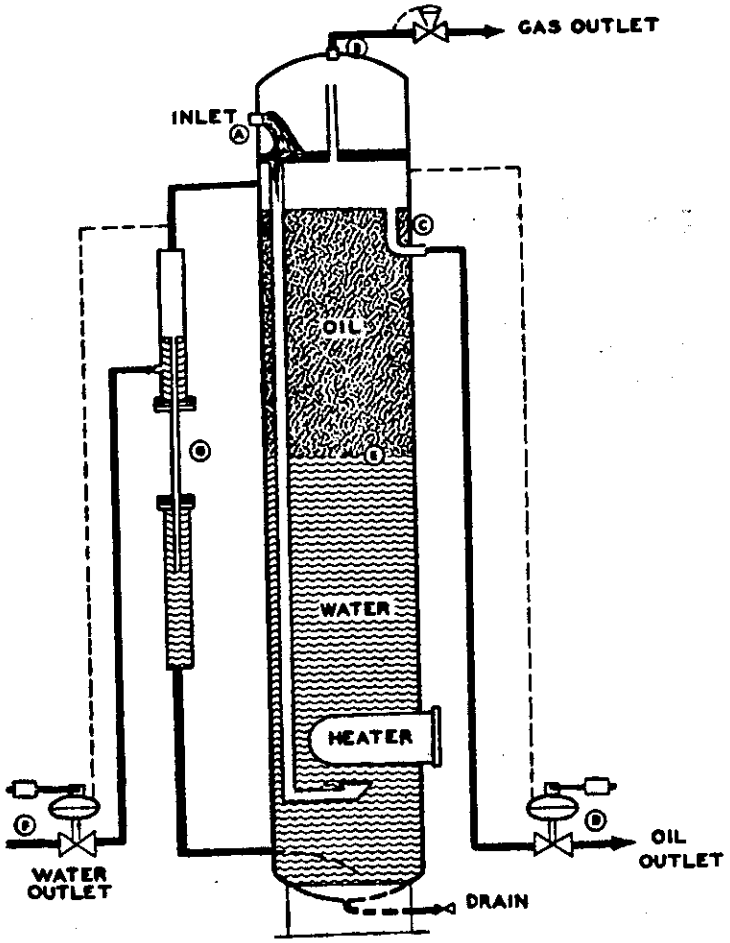
The control system shown with the high - pressure free - water knockout in Fig. 3 is similar to Fig. 1. In this case though a conventional liquid-level controller is incorporated into the water-outlet compartment.

The oil simply overflows into the gas-outlet line with the combined stream passing through the gas back-pressure valve. Because of liquid surging the controlling connection should be upstream from where streams commingle.

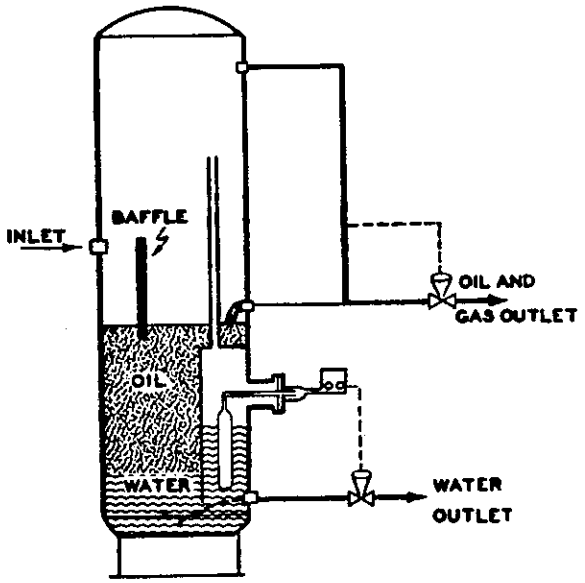
Vessel Sizing

Any liquid-liquid separation vessel must be sized to accommodate the gas load without entrainment and have sufficient cross-section area and liquid volume so that the heavier fluid can settle out in the time it is retained in the vessel.

It was once felt that entrainment greater than that from separators could be tolerated but experience has shown this to be undesirable. Consequently the gas capacity should be determined in the same way as for gas separators.



TREATER SYSTEM of water and oil separation is similar to the free-water knockout (Fig. 1) but a valve is provided for control of the discharge of both oil and water. Fig. 2.



HIGH-PRESSURE KNOCKOUT of free water has liquid-level control on water outlet. The oil overflows into the gas outlet line. Fig. 3.

The sizing for liquid capacity is considerably more of an art than a science. The best approach makes use of basic force concepts and the application of Stokes Law to a free-settling particle.

The force causing a particle to settle is:

$$F = m_w g - m_o g - F_d \quad (3)$$

since:

$$F = m_w (dv/dt) \quad (4)$$

$$m_w (dv/dt) = m_w g - m_o g - F_d \quad (5)$$

From this equation one may obtain:

$$m_w\,[(\rho_w - \rho_o)/\rho_w]\,g = F_d \qquad (6)$$

if it is remembered that:

$m_o = \rho_o \times$ oil volume.

$m_w = \rho_w \times$ water volume.

Since the volume of oil displaced must equal the droplet volume:

$$m_o/m_w = \rho_o/\rho_w$$

When the droplet has reached an equilibrium velocity:

$$dv/dt = 0$$

From Stokes Law:

$$F_d = 3\pi\,\mu\,v\,D_p \qquad (7)$$

Combining Equations 6 and 7 yields:

$$v = \frac{g\,D_p^2\,(\rho_w - \rho_o)}{18\,\mu} \qquad (8)$$

if the droplet is assumed to be spherical:

$$m_w = (\pi\,D_p^3/6)\,\rho_w$$

The equation is sound enough but in the actual vessel absolute free settling cannot occur. Furthermore the particle size D_p not only varies but is indeterminable. The path of the droplet also takes it past fluid of varying density. Consequently the use of Equation 8 involves the use of several assumptions.

The velocity obtained from Equation 8 may be used to determine vessel diameter by the equation:

$$D_e = [(0.000083)\,Q/v]^{1/2} \qquad (9)$$

Where both gas and liquid are separated in the same vessel as shown in Fig. 3 the necessary diameter would be one that would satisfy Equation 9 and still handle the gas load.

Nomenclature

D_e = vessel diameter to handle liquid, ft.
D_p = diameter of water droplet, ft.
F = force of motion of particle, poundals
F_d = resisting force to motion, poundals
g = gravitational acceleration, ft. per second^2
m_w = mass of water, lb. mass
m_o = mass of oil, lb. mass
Q = water load, bbl. per day
v = droplet settling velocity, ft. per second
t = time, seconds
π = 3.1416
ρ_w = density water, lb. per cu. ft.
ρ_o = density oil, lb. per cu. ft.
μ = viscosity, lb. per ft.-second

ELEMENTS OF FIELD PROCESSING—21

TEMPERATURE CONTROL—1

by Dr. John M. Campbell

THE heaters used in oil production operations may be classified into several categories:

1. Heated fluid passes through heater.

A. Indirect type, using: (a) water, (b) heat-transfer oil, (c) eutectic salt.

B. Direct type.

2. Heating medium circulates

A. Hot water

B. Low-pressure steam.

C. Heat-transfer oil, Dowtherm, etc.

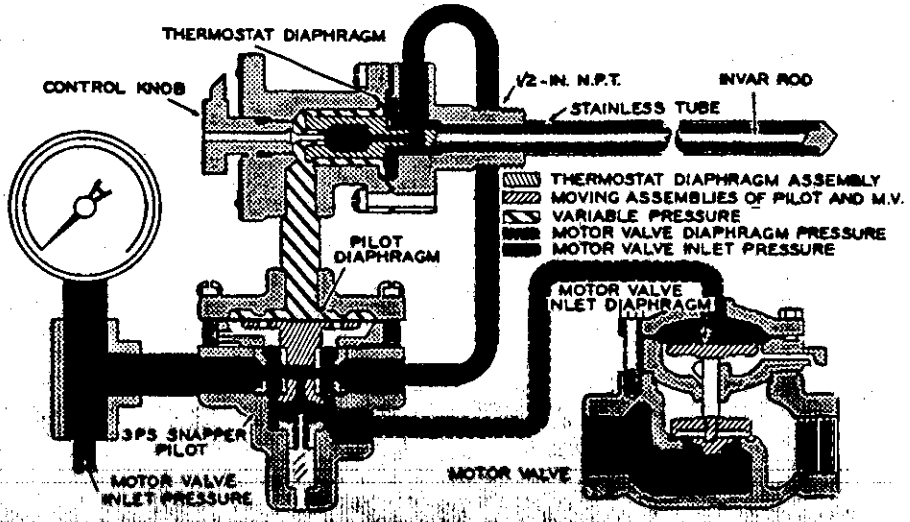
An indirect heater is basically a device whereby the burner heats a fluid bath which in turn heats the fluid passing through the flow coils. In this manner the hot flame is isolated from the hydrocarbon which minimizes coking and burnouts and increases the ultimate safety. The choice of fluid bath is of course a function of the temperature range desired.

From a control standpoint the large mass of the liquid bath used complicates the problem, particularly if accurate temperatures are required. On oil and gas lines where only approximate temperatures are needed it is usually sufficient to control the bath temperature. However, where the heat output must vary, a controller must be placed on the flow coil, for the bath acts as a "heat sponge" and prevents rapid response to the temperature pilot.

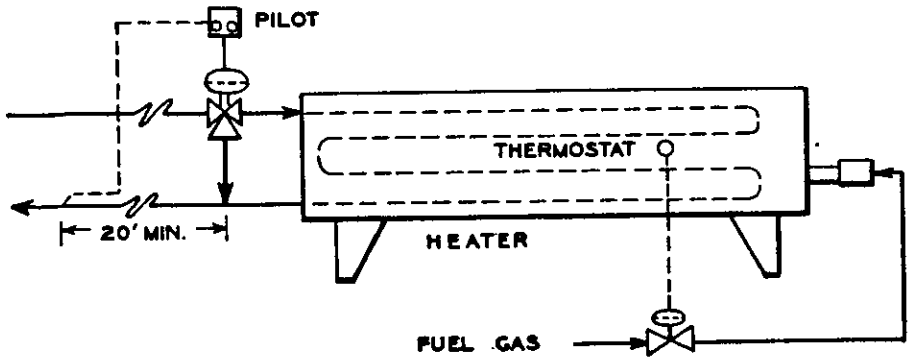
On-off control . . . Contrary to the usual situation, the simplest instrument is usually the most desirable for control of an indirect heater. In fact not over 10 per cent proportional control is recommended because the bath temperature would always lag behind the controller. This could be overcome by using an instrument with derivative response but the cost would not be justifiable. In this application reset would not be needed in any event.

A thermostat of the type shown in Fig. 1 is very satisfactory in this service. The probe mounted in the vessel consists of an invar rod within a stainless-steel tube. Changes in temperature will change the length of the tube but have very little effect on the invar rod. The relative movement of these two elements then controls the action at pilot seats (1) and (2).

In the particular model shown the thermostat action then actuates a snap-acting pilot which in turn operates a



CONTROL OF AN INDIRECT HEATER can be accomplished very satisfactorily with a thermostat of this type. Fig. 1.



THIS CONTROL SYSTEM, containing reset and derivative responses, is necessary for some heater installations. Fig. 2.

simple diaphragm valve. Using this means of control the burner is either on or off. From the control standpoint alone this is desirable for it enables the fast transference of heat to the relative large mass represented by the liquid bath. On large heaters, though, the large surge of gas when the valve opens is sometimes felt to be undesirable and some degree of throttling is needed.

Varying flow rate . . . When a control is needed on the fluid through the coil the thermostat shown in Fig. 1 is not entirely satisfactory, particularly if the flow rate varies widely. It may give good results in cases where the loading is relatively constant, when equipped with a proportional pilot.

It has been found necessary in many cases to add reset in order to hold the temperature at the control point and derivative response to compensate for the time lag inherent in the system. Fig. 2 is an example of this type of control. The thermostat holds the bath temperature at the predetermined level, preferably greater than 50° F. above the desired outlet temperature. The three-way valve then bypasses enough of the cool fluid to give the desired outlet temperature. Under design conditions some fluid should be bypassed so that control in both directions is provided.

The temperature sensing bulb should be at least 20 ft. from the bypass point to allow mixing of the streams. This is an inherent disadvantage of the system for it introduces a time lag that often must be corrected by derivative response.

The three-way valve may be replaced by a less-expensive two-way valve located on the bypass. This valve must be carefully sized, for the pressure drop is small through the heater coil and consequently it has to operate on a small pressure differential. In the final analysis the three-way valve is usually worth the extra investment because of the flexibility it affords.

Direct-Fired Heaters

In direct-fired heaters no buffer solution is used between the fire and the flow tubes, the heat being transferred by direct flame impingement, radiation from the flame, and the hot exhaust gases. They offer the advantage of very positive heat transfer since the mass involved is relatively small. Their chief disadvantage stems from the large temperature differential that exists across the tube wall which increases the danger of hydrocarbon coking, fouling, and ultimate failure of the tubes. It is for these reasons that they have been used largely in plants where they could be supervised and high heat loads per unit volume are encountered.

The basic system usually consists of a temperature pilot on the fluid outlet that controls the gas to the burner. In the event that this control fails, a high-temperature shutoff is used to prevent heater damage. A proportional pilot customarily is used, for it gives good control and minimizes thermal stresses in the flow coil. The manufacturers of tubing discourage the use of on-off control, for repeated heating and cooling sets up stresses that can ultimately cause failure. Recent metallurgical developments, however, have minimized this problem.

The need for derivative response would normally be less than with the indirect type but reset would still be required for varying flow rates.

ELEMENTS OF FIELD PROCESSING—22

TEMPERATURE CONTROL—2

by Dr. John M. Campbell

IT is often desirable to locate the burner away from the heat-transfer equipment. Such an installation is more expensive than the use of indirect heaters but may often be justified from considerations of safety and flexibility. The primary use is therefore in tank heaters, treaters, and stabilizers where these features are desirable.

Where the temperatures needed are not above 180° to 190° F. the 15-psig. steam generator is an efficient and flexible source of heat. The use of steam on the lease at pressures higher than 15 psig. is limited by present statutes and boiler codes which prohibit such vessels without continuous supervision.

The typical steam generator is gas-fired and has a thermostat and pressurestat as primary temperature controls. The thermostat is used as the primary control with the latter shutting off the burner supply gas if the pressure gets too high. A self-operated level-control valve is used for water makeup if such is available in the field. Because of the low water loss from the closed system a manual makeup is often used. The condensate normally flows back to the steam generator from the heat-transfer coil by gravity and no control is necessary.

Steam controller . . . The heating system shown in Part A of Fig. 1 is simple and transfers a large amount of heat per unit area of coil surface. In most field applications it is difficult to use a simple self-actuated temperature controller that has at least a 10 per cent proportional band. The valve may be located in either position (1) or (2) as shown.

Position (1) is preferred, for it requires a smaller valve and will usually give slightly more accurate control. In this position the temperature in the vessel is controlled by the height of condensate in the coil, i.e., changing the area available for transfering heat to the vessel. For example, if the temperature is too low the valve opens wider to drop the level and expose more area. In the event that the heating element is a low coil, position (2) would be necessary.

Hot liquid . . . A typical hot-liquid circulation system is shown in Part B. The heater may be of any type but it is often a low-pressure modified steam generator. In this service it preferably has two thermostats, one for control and one as a high-temperature shutdown. The working pressure is usually 40 psig.

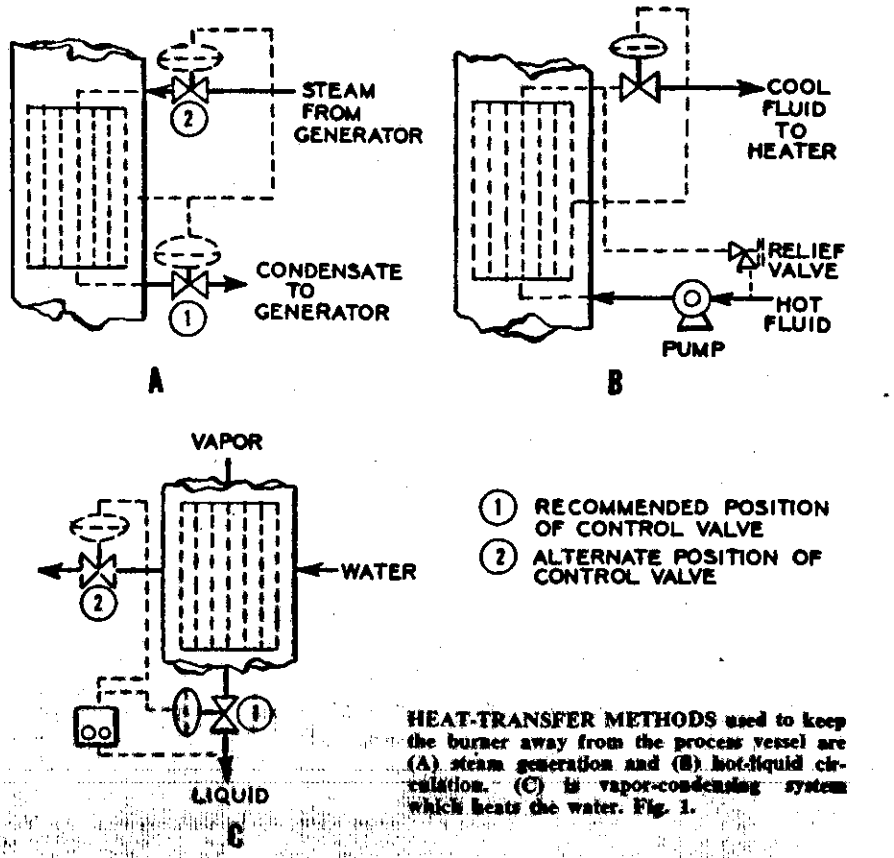
On the usual lease system hot water, circulated with a centrifugal pump, is used as the heating medium. For this service the control valve would be placed on the coil outlet. The bypass with the relief valve is not essential but often desirable when an open-impeller type of pump is used.

With a close-clearance centrifugal or positive-displacement pump the bypass is essential to keep from overpressuring the system if the control valve closes. When using the latter, the control valve may be placed on the bypass. The proper amount of fluid is then bypassed around the heater to maintain the proper temperature. The circulation rate should be such that this valve will be normally half open, affording temperature control in both directions. If the pressure drop across the heating coil is small the system shown in Part B is preferred, for a valve on the bypass would have to operate on too small a pressure differential.

Hot fluid may be circulated for services such as tank heating by means of thermo-syphon and gas jet action to eliminate the pump. The former makes use of the differential densities of hot and cold water and is not a very positive or flexible system of heating. The gas jet is more positive but finds decreasing application because of the increasing value of gas. Very few such systems are still seen, for tank heaters have been largely supplanted by compact treaters and heater-treaters.

Condenser . . . Part C shows two accepted modifications on a system where a vapor is being condensed, using cold water. The amount of heat transferred may be fixed by controlling the amount of water circulated or by fixing the flow rate and varying the heat transfer surface available. Where the heating coil has sufficient height, the latter method is preferred. The water rate may then be controlled manually.

In most oil-field applications there is no significant difference in the two methods. When using stacked finned-tube heat exchangers the valve must be placed in position (2).



HEAT-TRANSFER METHODS used to keep the burner away from the process vessel are (A) steam generation and (B) hot-liquid circulation. (C) is vapor-condensing system which heats the water. Fig. 1.

ELEMENTS OF FIELD PROCESSING—23

Control of Low-Temperature Systems

by
Dr. John M. Campbell

CONTROL of expansion type of low-temperature separation systems represents a special problem, particularly where hydrates are encountered. Unfortunately no one satisfactory method of control is applicable for all systems due to variations in pressure, fluid properties, and the system used.

Two typical low-temperature systems are shown in Figs. 1 and 2. Both represent the type of system where hydrates are allowed to form and then melted. Fig. 1 is typical of that system for high-pressure, high-temperature making salt water while Fig. 2 is primarily for small wells, making no salt water, which do not justify a high equipment investment. The two systems shown represent extremes with many variations in between.

Preventing hydrates . . . The problem in all cases is to prevent hydrate formation prior to the choke and yet supply enough heat to melt the hydrates formed when the gas expands across the choke. Wherever possible it is desirable to furnish this heat from the well stream without lowering the temperature below the hydrate point. The following general variations are encountered:

1. The well stream is too cool and a heater is used to prevent hydrate formation in the flow lines and supply heat for the separator coil. (Fig. 2).

2. A low-pressure steam generator is used to supply heat to the separator coil, the well stream flowing directly to the choke.

3. Heat for the separator coil is furnished from the well stream as shown in Fig. 1.

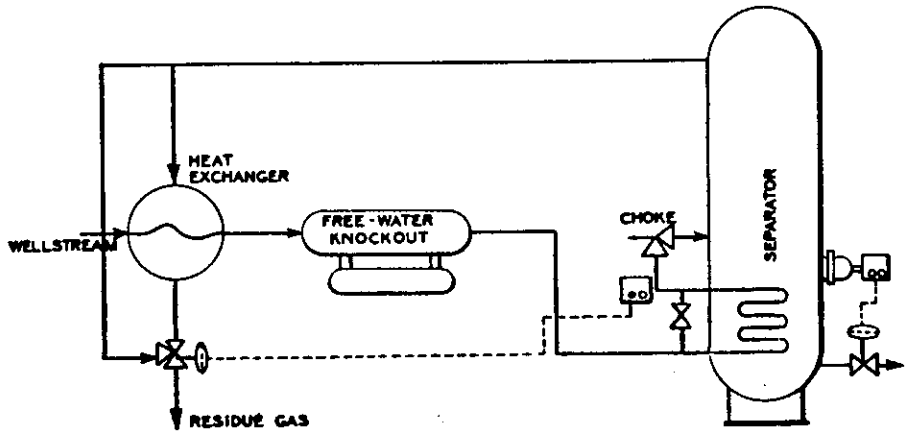
The gas-gas heat enchanger is added for Method 3 above when the well stream contains a surplus of heat. This is normally encountered on production from deep horizons as typified by Gulf Coast operations.

The free-water knockout is an essential item in the large majority of wells for even relatively small amounts of free water in the flow lines and around the choke are a potential source of trouble. There have been some applications, however, particularly in Oklahoma and Kansas, where salt water has been no trouble.

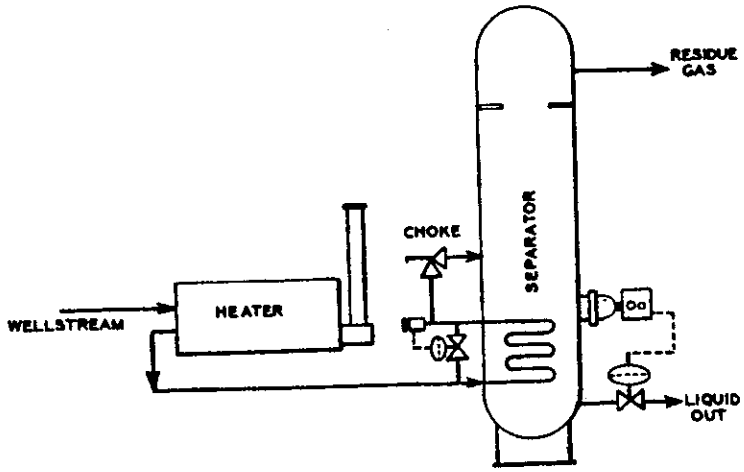
Using well stream . . . The system shown in Fig. 1 is a simple yet efficient means of control. In most systems it is desirable to carry the gas temperature entering the choke at about 5° to 6° F. above the hydrate temperature. The only means of positively controlling this temperature is by either (1) bypassing the heat exchanger or (2) bypassing the separator coil, either of which will raise the gas temperature. Either approach is possible but the former is the more sensitive. In any event a bypass around the coil should be provided in the event of malfunction or should the coil freeze.

Inasmuch as the desired gas temperature is usually close to the hydrate point, accurate control is necessary. Variations in flow rate, atmospheric temperature, etc., will all affect this temperature. Consequently, proportional control plus reset is advisable, for without reset the desired temperature will drift. Most manufacturers of this equipment furnish only thermostatic control for competitive reasons. The author has often noted, however, that where trouble is encountered for this reason the pumper simply raises the control temperature to where the normal drift causes no trouble. This is indeed an expensive way to save money on control equipment.

A two-way valve located on the gas bypass could be used instead of the



SURPLUS HEAT in a high-pressure well stream is removed by the gas-gas heat exchanger and the separator coil. Here control of the gas temperature to prevent hydrate formation at the choke is by bypassing the exchanger. Fig. 1.



COOL WELL STREAM is passed through heater, then through separator coil. Heat removed in the coil melts hydrates formed by gas expansion across choke. Coil bypass prevents too low gas temperature. Fig. 2.

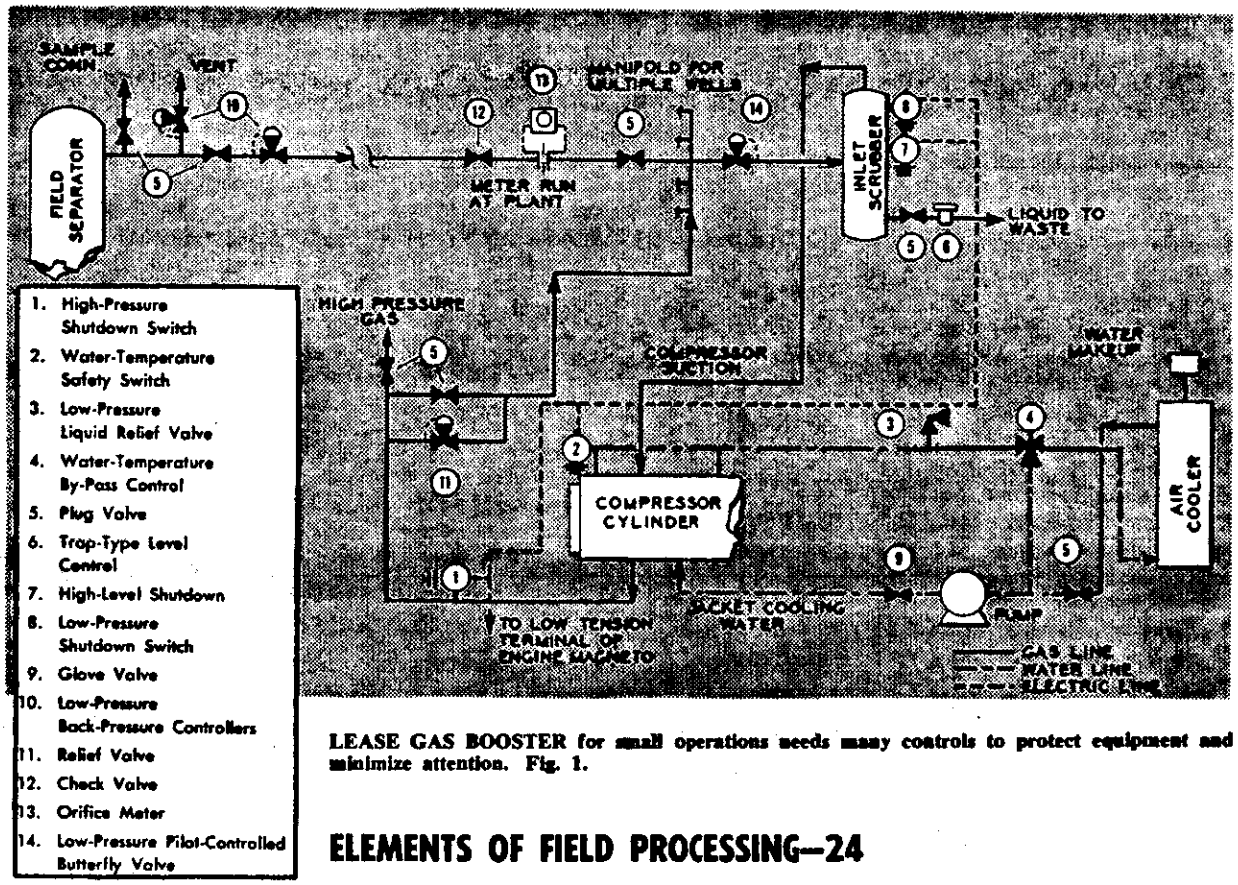
three-way valve shown if the pressure drop across the heat exchanger is high enough so that the valve may operate properly. This results in a cost reduction but once again the control is not quite so positive. If a three-way valve is used it should be on the outlet rather than the inlet side of the heat exchanger.

Use of heater . . . The simpler system shown in Fig. 2 has correspondingly simpler controls. The temperature of the gas to the coil is set by the thermo stats in the water bath of the indirect heater. The control on the separator-coil bypass is not essential but is a desirable addition. The heater control is "rough" and there are periodic changes in the coil heat demands. This bypass therefore supplies the needed flexibility.

The thermostat is normally set so that the bypass valve is fully open when the gas leaves the coil about 5° to 6° F. above the hydrate point. The heater is then adjusted so that the fluid leaving the separator is not more than 80° F. with the bypass valve fully closed. In this manner the bypass is normally closed during operation and the thermostat serves primarily as a safety control.

An alternate control is provided by placing the thermostatic element in the body of the separator near the liquid outlet. The bypass is then used to control the liquid temperature. Since this desired temperature is nearly coincident with the usual hydrate conditions encountered, it automatically provides for a gas temperature high enough to prevent freezing. This method prevents too high a heater temperature from giving excessive reentrainment in the separator, but is inherently less sensitive.

The level control is standard and the same as that normally specified in Fig. 1.



LEASE GAS BOOSTER for small operations needs many controls to protect equipment and minimize attention. Fig. 1.

ELEMENTS OF FIELD PROCESSING—24

Control of Field Booster Compressors

by Dr. John M. Campbell
University of Oklahoma

MOST of the controls on a field booster compressor are designed to protect the equipment and facilitate operation, with minimum attention, for it is theoretically possible to operate the system with little more than manual control. The exact scheme used will of course vary with the application but that shown in Fig. 1 is typical of small lease operations where low-pressure gas is compressed in one stage.

No control and cooling system is shown for the engine, since this will depend on the engine manufacturer. On most small portable units the engine is self-contained with its own radiator and pump. If two-stage, or more, compression is used, the controls shown for the inlet scrubber would be duplicated for each interstage scrubber. A provision would also have to be made

for interstage gas cooling.

The back-pressure controller shown on the vent is to prevent overpressuring of the separator in case the compressor plant is down. The other back-pressure valve on the suction line is to prevent the plant from lowering the separator pressure by drawing off gas faster than it is produced. It should be set several psi. lower than the valve on the vent line.

The meter run is shown at the plant although it could be conveniently located at the separator. Check valve (12) is included to prevent the back flow which may result when multiple units feed the same compressor.

Butterfly valve (14) is an optional but desirable item to control the compressor suction pressure. All compressors are rated using a given suction and discharge pressure and this valve simply prevents the former from getting too low and overloading the unit. This valve is particularly desirable if the deliverability of gas to the plant is widely variant.

The inlet scrubber is primarily a protective device to keep liquid out of the cylinders and should not be used as a primary oil and gas separator. This vessel serves simply as a liquid sump and a convenient device on which to hang certain necessary controls. As a convenient rule-of-thumb, this vessel should have a cross-sectional area of at least 100 sq. ft. per M.M.c.f. per day of gas processed. The minimum size is normally 12 in. A float-operated trap (6) is very satisfactory for controlling the liquid level.

Switches (1), (2), (7), and (8) constitute the basic safety shutdown system. These switches are normally open but if the condition at any one goes beyond set limits it closes, grounding out the engine magneto, which in turn stops the engine. All are simple and inexpensive, and afford positive protection. A horn or remote indicator may be incorporated that is actuated any time the compressor stops.

Switches (1) and (8) are high and low-pressure shutdowns respectively. They serve to keep these pressures within the limits set up for the compressor. Switch (2) is a high-temperature shutdown that is actuated by the hot water leaving the cylinder jacket. It shuts down the unit when any failure occurs in the cooling system.

Switch (8) is a high-level shutdown that operates when the liquid level becomes too high in the scrubber due either to failure of valve (6) or to an excess quantity of liquid entering the plant.

Bypass valve (5) downstream from the compressor is desirable for it provides a means of taking the load off the compressor, particularly during startup. During normal startup this valve would be open and gradually closed once the compressor and engine reached running speed.

The bypass relief valve (11) is an optional and, to a certain extent, a luxury item for it duplicates the protection offered by switch (1). It provides relief in the event the outlet or bypass valves are inadvertently closed, without shutting down the compressor. If used it would normally be set to open 5 to 10 psig. before the high-pressure switch closes.

The water-temperature bypass control (4) serves to control the jacket-water temperature by governing the amount of water to the cooler. In its simplest and cheapest form it consists of an automobile-radiator type of thermostat in a three-way cast-iron valve body. It is substantially the same device used on most oil-field engines. A conventional temperature control would of course be satisfactory but its cost cannot be justified unless it is a large installation or where the complexity of the system does not lend itself to this type of control. In some cases where several services are handled by the same cooler it is often more satisfactory to control the temperature by varying the speed or pitch of the fan.

Although no scale trap is shown it is normally a good policy to have one to prevent accumulation and possible system damage. Some provision for water makeup is necessary so a small surge tank on top of the cooler should be provided for this purpose. Drains should be provided so that the system may be completely drained.

Popoff valve (3) is primarily to prevent overpressure of the system, particularly the radiator. A simple brass valve ½ to 1 in. in size is normally sufficient.

On large systems a standpipe may be used ahead of the pump suction to handle surges and serve as a water reservoir.

"It's a good policy to have a scale trap to prevent accumulation and possible system damage."

ELEMENTS OF FIELD PROCESSING—25

Relief and Venting Equipment

by Dr. John M. Campbell

THE installation of proper relief and venting equipment is an integral part of the control system. No vessel should ever be installed, regardless of the pressure range, without provision for the adequate relief of excess pressure.

The specifications governing such relief equipment are normally set up under one or more of the following codes:

1. A.S.M.E. Unfired Pressure-Vessel Code.
2. A.S.M.E. Power Boiler Code.
3. A.P.I.-A.S.M.E. Unfired Pressure-Vessel Code.

With the possible exception of L.P.G. systems and steam generators most lease equipment is built and installed under the last code. This code provides that all unfired vessels shall be protected by a pressure-relieving device that will prevent the pressure from rising more than 10 per cent above the maximum allowable working pressure stamped on the vessel.

The pressure-relieving device may be a relief valve and/or a rupture disk. The former is ordinarily a direct-acting spring-loaded valve except that a pilot-loaded valve may be used if it is of the normally open type and will discharge its full rated capacity when the pilot fails. This spring-loaded valve, though, offers economic advantage and is dependable.

... **The rupture disk** is carefully made in such thickness and shape that it will burst at a given pressure ±5 per cent. These disks have the advantage of being very positive, providing a tight seal and being available in many special metals at moderate cost. In most sizes the disk is supported between a special flange but small union type and threaded "throwaway" models are available. The primary disadvantage is that it cannot seal off again after relief of pressure.

Both a relief valve and rupture disk may be used in parallel, in which case the valve is usually set for 10 per cent overpressure and the rupture disk 50 per cent. A rupture disk may also be connected in series with the valve, either before or after, provided certain special code stipulations are met.

A rupture disk may be placed ahead of the relief valve to prevent leakage or to isolate the valve from viscous or corrosive materials. In this case the disk should have a slightly higher bursting pressure than the valve set pressure. It is necessary to have a connection between the two to check the intactness of the disk and enough room so that the disk may rupture completely. In fact some provision to prevent it from fouling the valve is often indicated.

On certain systems the rupture disk may be placed on the valve outlet, the sole function being to prevent leakage of valuable or noxious materials. If this is done a back pressure is superimposed and a bellows type of valve or equivalent, discussed below, should be used.

Rupture limit . . . Regardless of the use, the rupture disk should have a bursting pressure at least 50 per cent above the maximum normal operating pressure for the best service. This can be a serious limitation where the code specifies that the vessel pressure shall not exceed the vessel working pressure by over 10 per cent. Unless some other arrangement is negotiated with the insuring company this means that the vessel's working pressure has to be 1.36 times the anticipated operating pressure if the above two conditions are strictly adhered to. This certainly limits the application.

In choosing the material of construction it is also necessary to consider the temperature at which the disk will operate and the materials with which it will come in contact. At elevated temperatures metals lose their strength while at low ones embrittlement is a problem. Fortunately many alloys are available to minimize the problem.

Precautions . . . When installing a relief system the following general provisions should be considered:

1. Vessels completely filled with liquid should have a liquid relief valve with a diameter not less than that of ½-in. pipe.

2. No obstruction should be present between the vessel and the relieving device and the pipe should be at least as large as the valve inlet.

3. When two or more pressure-relief devices are connected to the same opening, said connection should have a cross-sectional area equal to the sum of the areas of the device inlets.

4. The spring in a relief valve should not be reset for any pressure that deviates more than 10 per cent from the specified valve pressure when the total pressure does not exceed 250 psig. Above 250 psig. only a 5 per cent deviation is allowable.

5. When two or more vessels are connected together by adequate piping, without valves, so that the vessels may not be isolated, these vessels may be considered as one unit in calculating relief requirements.

6. Where an additional hazard such as fire is present, as in hydrocarbon storage, supplementary relief devices are necessary. They must be sized so that the pressure shall not rise over 20 per cent above the vessel's maximum allowable working pressure.

7. All relief valves should open at the maximum allowable working pressure of the vessel.

8. A plug valve may be placed between the relief valve and the vessel provided: (a) that said valve will not reduce the relieving capacity, (b) that the valve is sealed open and is closed by an authorized person only where repairs or other work demand it.

9. Any vent system downstream from the relieving system should be designed to minimize the effect of variable back pressure, contain no plug valves except as provided in (8) and be designed to facilitate drainage.

10. The pressure drop in discharge lines should ideally not exceed 10 per cent of the set pressure. One common practice is to size this line so that its area is equal to that of all the valve outlets discharging into it.

Back pressure . . . Two types of back pressure are encountered in vent systems, viz., "built up" and "superimposed." The former is the type of pressure that builds up after the valve opens, particularly where the lines are long. A superimposed back pressure is one that exists by virtue of deliberately holding a pressure on the system or from the prior opening of other valves. In either case the back pressure must be dissipated from (or prevented from entering) the bonnet since this force would act as an additive force to the spring and prevent the valve from opening at set pressure.

One manufacturer minimizes this difficulty by using an eductor tube which keeps the bonnet pressure low when only "builtup" pressure is encountered. However, where the superimposed back pressure is high or variable it is necessary to keep this pressure from the bonnet and the working parts. The best way of accomplishing this is probably to have a bellows that seals off the bonnet and working parts from the outlet fluid. This bellows also serves to protect against corrosion, viscous fluids, and contamination.

ELEMENTS OF FIELD PROCESSING—26

Sizing of Relief Equipment

by Dr. John M. Campbell

THE capacity of any relief device is dependent on the following major variables: (1) Orifice opening, (2) specific gravity of fluid, (3) temperature of inlet gas, (4) compressibility factor, (5) a nozzle coefficient, (6) absolute flowing pressure (including allowable overpressure), and (7) back pressure on the relief device. The following equations are therefore proposed by the A.S.M.E. and A.P.I. for calculating the necessary opening for a relief valve:

Vapors:

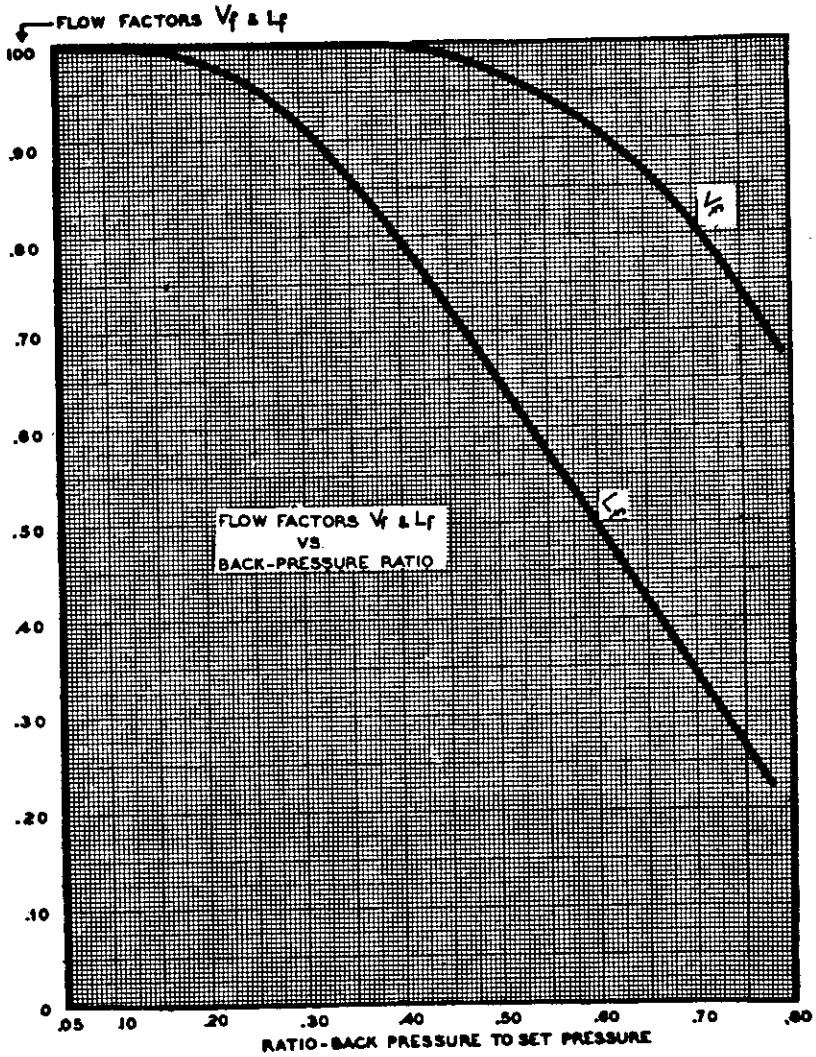
$$A = \frac{Q_v (STZ)^{1/2}}{1.175\ V_f\ C_1\ P_1} \qquad (1)$$

Liquids:

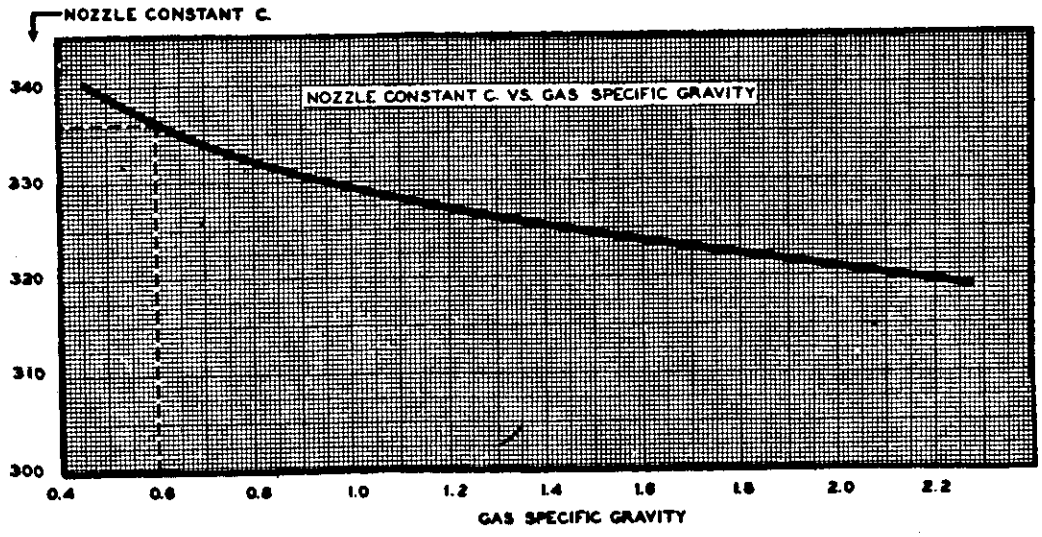
$$A = \frac{Q_L (S)^{1/2}}{27.2\ L_f (P)^{1/2}} \qquad (2)$$

Where:

A = orifice area, sq. in.
Q_v = gas capacity, s.c.f. per minute
Q_L = liquid capacity, g.p.m.
S = specific gravity of fluid, dimensionless
P = differential pressure (set pressure − back pressure)
P_1 = flowing pressure (set pressure + allowable overpressure and barometric pressure), psia.
T = inlet temperature, °R.
Z = compressibility factor, dimensionless



FLOW FACTORS for vessels with a working pressure greater than 100 psig. Fig. 2.



NOZZLE CONSTANT, C, is a function of nozzle-flow coefficient and gas specific gravity. Here, nozzle-flow coefficient is taken as 0.975. Fig. 1.

C_1 = nozzle coefficient, consisient units
V_f = gas flow factor, dimensionless
L_f = liquid flow factor, dimensionless

The nozzle-flow factor C_1 is obtained by combining the gas constant shown in the A.P.I.-A.S.M.E. and A.S.M.E. codes with the nozzle-flow coefficient of the specific valve. This coefficient is a function of the ratio of specific heats of the gas (C_p/C_v) which in turn may be correlated against specific gravity. Fig. 1 was prepared using this relationship and a nozzle-flow coefficient of 97.5 per cent.

Where the back pressure is less than 20 per cent of the set pressure, V_f and L_f may be considered equal to one. Above this pressure ratio only a bellows type of valve should be used (see Part 25). Fig. 2 summarizes the values of V_f and L_f for vessels with a working pressure greater than 100 psig.

With these figures plus knowledge of the other variables in Equations 1 and 2, a value for "A" is readily obtained. The orifice sizes for a valve are designated by the letters shown in Table 1. The valve chosen therefore should have an orifice letter corresponding to the next larger orifice area than the one calculated.

In choosing a specific valve it is further necessary to consider the pressure rating of the body and the inlet and outlet flanges, the inlet temperature, materials of construction, and any special limitations such as back pressure, etc. These will be outlined in the manufacturer's catalog. In some cases convenient nomographic solutions of Equations 1 and 2 are also available.

Example: A relief valve is to be installed on a gas separator that has a working pressure of 500 psig. at 100° F. If the vessel has a capacity of 10 M.M.c.f. per day of 0.60-sp. gr. gas, determine the proper orifice designation for the relief valve. The back pressure on the vent system is negligible.

Solution:

$$Q_v = \frac{10 \times 10^6}{(24)(60)} = 6{,}940 \text{ s.c.f. per minute}$$

$V_f = 1$, $C_1 = 336$ (Fig. 1),

$T = 560°$ R.

$P_1 = 500 + 14.7 + (0.10)(500) = 564.7$ psia.

From established specific gravity correlations, $Z = 0.93$.

From Equation 1:

TABLE 1—NOZZLE ORIFICE AREAS

Orifice designation	Orifice area (sq. in.)
D	0.110
E	0.196
F	0.307
G	0.503
H	0.785
J	1.287
K	1.838
L	2.853
M	3.60
N	4.34
P	6.38
Q	11.05
R or S	16.00
T	26.00

$$A = \frac{(6{,}940)\,[(0.6)\,(560)\,(0.93)]^{1/2}}{(1.175)\,(1.0)\,(336)\,(564.7)} = 0.55 \text{ sq. in.}$$

Therefore the valve designated should have an H orifice (Table 1) since this has the smallest area satisfying the above result.

The above type of relationship may be used to calculate the flow capacity of a rupture disk since it also involves a modified orifice relationship. The coefficient, however, would depend on the fracture pattern established by the disk when rupturing and the degree to which the remaining metal hindered flow. For these reasons manufacturers' curves should be consulted.

ELEMENTS OF FIELD PROCESSING—27

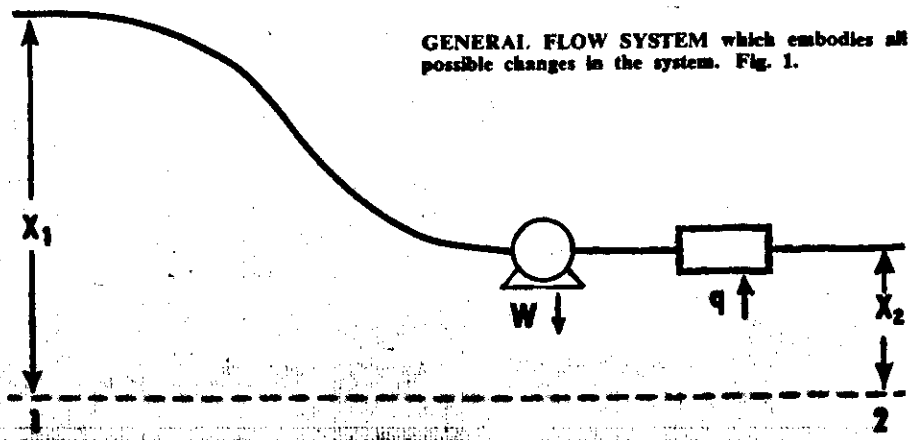
Derivation of Flow Equation for Liquids

by Dr. John M. Campbell

A SATISFACTORY equation for the prediction of fluid-flow characteristics is best derived from the first law of thermodynamics which in general terms states the indestructibility of mass and energy, or their equivalent. This then enables one to write an energy balance around any two points in a flow system.

Such a general system is shown in Fig. 1, which embodies all of the changes possible in the system. The balance may be written between any two points in the system but the general balance between points (1) and (2) would be:

$$U_1 + P_1V_1 + X_1 + \frac{V_1^2}{2g} + q = W$$



GENERAL FLOW SYSTEM which embodies all possible changes in the system. Fig. 1.

$$= U_2 + P_2V_2 + X_2 + \frac{v_2^2}{2g} \quad (1)$$

or $\Delta U + \Delta(PV) + \Delta X + \frac{v_2^2 - v_1^2}{2g} = q - w \quad (2)$

Where:

- U = internal energy which includes all energy present including heat, electrical, chemical, surface, etc.
- P = pressure
- V = volume or specific volume
- X = elevation above an arbitrary plane
- v = velocity in conduit
- g = gravitational constant
- q = heat added or lost by system (plus if gained, minus if lost)
- W = work done by system (plus if done by system, minus if done by surroundings on system)

The fluid entering at (1) has an internal energy U_1 which is the result of previous energy obtained. In addition a certain amount of PV work must be done to get the fluid to flow past point (1), i.e., the energy necessary for it to flow. This fluid also possesses potential energy X_1, above the datum plane, and kinetic energy $v_1^2/2g$.

The fluid leaving at point (2) possesses the same forms of energy which are different in amount from those at (1) by the amount of heat and work gained or lost. Equation 2 is a convenient form of the final balance since normal data available are relative.

Another form of Equation 2 is obtained by remembering that the enthalpy $H = U + PV$. Therefore:

$$\Delta H = \Delta U + \Delta(PV) \quad (3)$$

Substituting (3) into (2) yields:

$$\Delta H + \Delta X + \frac{v_2^2 - v_1^2}{2g} = q - W \quad (4)$$

which is a convenient form for use when enthalpy data are available.

If chemical, electrical and surface energy changes are assumed negligible in the pipeline:

$$\Delta U = \int TdS - \int PdV \quad (5)$$

where S is the entropy and T the absolute temperature. Furthermore:

$$q = \int TdS - W_f \quad (6)$$

where W_f is the total of the irreversible effects present in the flow system. The term $\Delta(PV)$ is also equivalent to $\int VdP + \int PdV$. Substituting these terms into Equation 2 yields:

$$\int VdP + \Delta X + \frac{v_2^2 - v_1^2}{2g} = - W_f - W \quad (7)$$

Equation 7 is as far as one can go using thermodynamics alone for it provides no way of solving for the term W_f.

Lost work . . . The total irreversible effects W_f are usually attributed to friction. The work done in overcoming friction through a distance dL is proportional to the surface in contact with the fluid, approximately proportional to the square of the velocity and proportional to the fluid density. By including a proportionality constant f, the equation may be written:

$$\text{frictional resistance} = (f)\,(dL)\,(\pi D)\,(v^2/2g)\,(\rho) \quad (8)$$

The weight of fluid in the pipe is the length dL multiplied by the cross-sectional area and the fluid density. Any frictional work would be represented by the frictional resistance moved through distance dL. Combining these concepts into Equation 8 then yields dW_f:

$$dW_f = \frac{(f)\,(dL)\,(\pi D)\,(v^2/2g)\,(\rho)\,(dL)}{(\pi/4\,D^2)\,(\rho)\,(dL)} \quad (9)$$

Simplifying:

$$dW_f = \frac{2fv^2dL}{gD} \quad (10)$$

which integrated between the limits of 0 to W_f and 0 to L gives:

$$W_f = \frac{2fLv^2}{gD} \quad (11)$$

Combination of Equations 7 and 11 then gives:

$$\int VdP + \Delta X + \frac{v_2^2 - v_1^2}{2g} = - \frac{2fLv^2}{gD} - W \quad (12)$$

which is the basic equation for liquid flow.

For a line of constant cross-sectional area, flowing substantially incompressible fluid, containing no pumps or turbines, Equation 12 would simplify to:

$$V(P_2 - P_1) = - \frac{2fLv^2}{gD} - \Delta X \quad (13)$$

since $V = 1/\rho$ and the change in velocity in negligible.

$$\frac{\Delta P}{\rho} = - \frac{\Delta P_f}{\rho} - \Delta X \quad (14)$$

Where:

$$\frac{\Delta P_f}{\rho} = \frac{2fLv^2}{gD} = \text{pressure drop due to friction} \quad (15)$$

Where a means of finding f is provided, Equations 14 and 15 yield a solution for the pressure drop accompanying a given flow rate.

Calculation of pump horsepower . . . For this case, Equation 12 could be simplified to $V(P_2 - P_1) = - W$ because ΔX and the velocity term would be substantially zero. The friction drop is incorporated into an efficiency term E. If Q_L is expressed in g.p.m. the final equation may be expressed as:

$$W\,(\text{horsepower}) = \frac{Q_L(P_2 - P_1)}{(33{,}000)\,(7.48)\,(E)} \quad (16)$$

The work is thermodynamically negative which makes the left side of the equation positive. This and all equations above have consistent units therefore care must be taken to assure that this condition is satisfied. In every case subscripts (1) and (2) refer to the initial and final conditions respectively in the direction of flow.

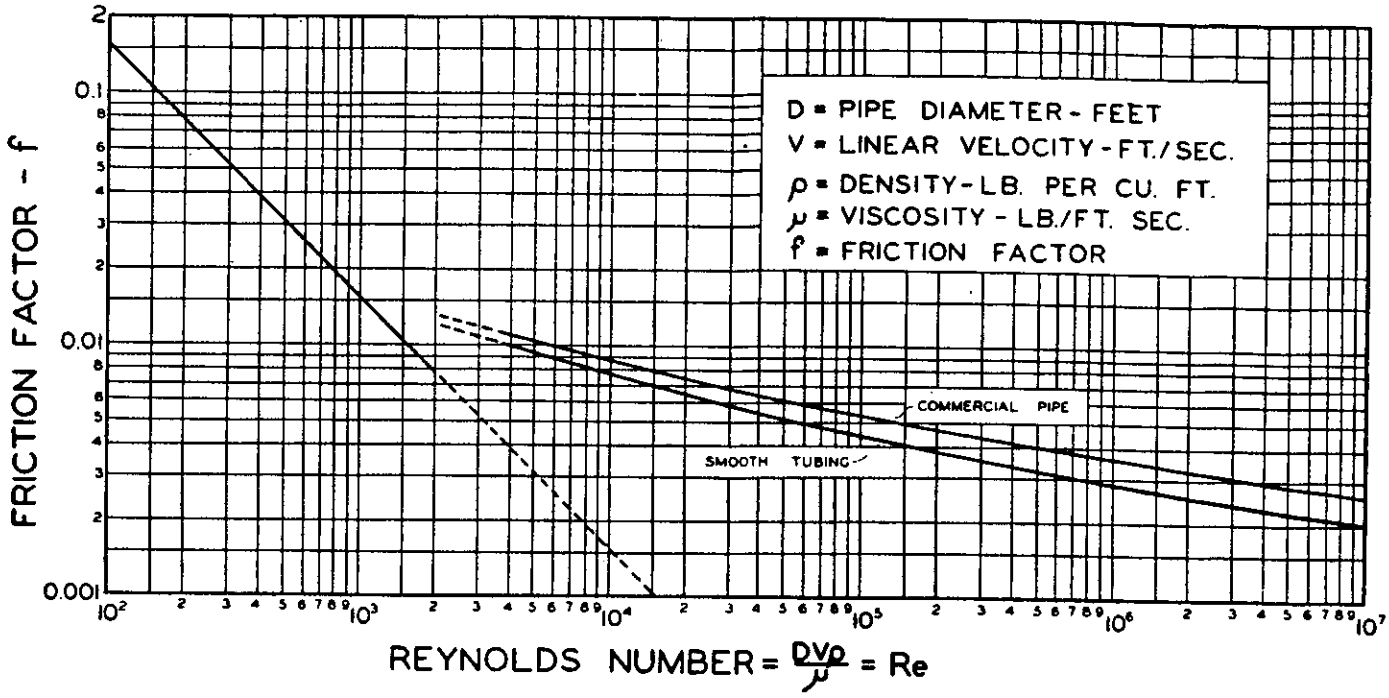


Fig. 1—Working chart for determining pressure drop.

ELEMENTS OF FIELD PROCESSING—28

Calculating Pressure Drop

by Dr. John M. Campbell

An equation:

$$V(P_2 - P_1) = -\frac{2fLv^2}{gD} - \Delta X$$

or

$$\Delta P/\rho = \Delta P_f/\rho - \Delta X$$

has been developed (Part 27) which enables one to calculate the pressure drop between any two points in a liquid system, said section containing no device to add or remove work. This equation may only be used, however, if the value f, called the friction factor, may be evaluated.

The application of dimensional analysis to experimental data has shown that f is a function of the dimensionless group Dvp/μ which is commonly known as Reynolds number or modulus (Re). The relationship between f and Re depends primarily on the roughness of the pipe wall and type of flow obtained. Although many combinations of these are possible the curves shown in Fig. 1 represent an excellent correlation.

Type of flow . . . Where Re is less than 1,000 the flow is laminar whereas above 4,000 it is turbulent. The zone in between is a transition region in which no good correlation exists but the use of the turbulent flow curves is recommended for they give conservative results.

Detail friction-factor curves are available which show the friction factor as a function of the pipe roughness. For most engineering uses, however, this degree of accuracy is somewhat misleading, particularly in lease processing work. The curves shown in Fig. 1, therefore, are recommended for those problems involving the calculation and design of gathering systems on oil leases and similar installations.

The approximate equations of these lines are:

$$Re < 1{,}000,\ f = 16/Re \qquad (1)$$

$$Re > 1{,}000,\ f = 0.04/Re^{0.194} \text{ for smooth tubing} \qquad (2)$$

$$Re > 1{,}000,\ f = 0.04/Re^{0.172} \text{ for commercial pipe} \qquad (3)$$

Although no exact criteria exist for differentiating between smooth tubing and commercial pipe, Equation 3 would normally be used for all new pipe 8 in. and less nominal diameter and all used pipe that had evidence of scale and corrosion.

Example . . . An 8-in. crude-oil line, 17 miles long, handles per day 40,000 bbl. of 40° A.P.I. oil with a viscosity of 9.5 cp., and discharges to atmospheric storage which is 300 ft. higher than the pump discharge. Determine the pump discharge pressure.

Solution:

$$40°\ A.P.I. = \frac{141.5}{131.5+40} = 0.824 \text{ sp. gr.}$$

$$= 51.4 \text{ lb./ft.}^3$$

$$9.5 \text{ cp.} = (9.5)(0.000672) = 0.00638 \text{ lb./ft.-sec.}$$

$$Q = \frac{(40{,}000)(5.61)}{(3{,}600)(24)} = 2.6 \text{ cu. ft./sec.}$$

$$D_{8"} = 8.08 \text{ in.} = 0.673 \text{ ft.}$$

$$v_{8"} = \frac{2.6}{0.355} = 7.32 \text{ ft./sec.}$$

$$Re = \frac{(0.673)(7.32)(51.4)}{0.00638}$$

$$= 3.98 \times 10^4$$

From Fig. 1, f = 0.0065.

Therefore:

$$\Delta P_f = \frac{2fLv^2\rho}{gD}$$

$$= \frac{(2)\,(0.0065)\,(17)\,(5{,}280)\,(7.32)^2\,(51.4)}{(32.2)\,(0.673)}$$

$$= 148{,}300 \text{ lb./ft.}^2$$

$$\Delta X = (51.4)\,(300) = 15{,}400 \text{ lb./ft.}^2$$

Therefore:

$$[(14.7)\,(144)] - P_1 = -148{,}300 - 15{,}400;$$

$$\text{or } P_1 = 148{,}300 + 15{,}400 + 2{,}120$$

$$= 165{,}820 \text{ lb./sq. ft. abs.} = 1{,}151 \text{ psia.}$$

$$= \mathbf{1{,}136\ psig.}$$

A convenient equation may be developed for oil lines using Equation 3 above. If B = oil flow rate in bbl./day and d = internal pipe diameter expressed in inches:

$$v = 0.286\ B/d^2$$

$$f = \frac{0.04}{Re^{0.172}}$$

$$= \frac{0.04}{(d/12 \times \rho/\mu \times 0.286\ B/d^2)^{0.172}} \quad (4)$$

Then:

$$\Delta P_f = \frac{2fLv^2\rho}{gD}$$

$$= 1{,}470\ \frac{LB^{1.828}\ \rho^{0.828}\ \mu^{0.172}}{d^{4.828}} \quad (5)$$

when the value of f from Equation 4 is substituted into the above equation and the result consolidated. The number of significant figures shown for the exponents result from the derivation and do not reflect the accuracy. For practical purposes these numbers may be rounded off to two significant figures.

Hazen-Williams Formula

The Hazen-Williams formula is an empirical equation that was originally developed for the flow of water. Like many such equations it contains an arbitrary constant for which a value is necessary. It has been used successfully for oil-line design where sufficient actual data were available to estimate a value of the constant C.

In common engineering terms this equation may be written as:

$$Q = 0.67\ CD^{2.63}\ (\Delta P/\rho L)^{0.54} \quad (17)$$

Where:

Q = flow rate, ft.^{3}/sec.
D = diameter, ft.
ΔP = pressure drop, lb./ft.^{2}
L = length, ft.
ρ = density, lb./ft.^{3}

The arbitrary constant C is primarily dependent on surface roughness but is also affected by pressure drop and diameter for the exponents used for these values usually differ slightly from the actual ones obtained.

Saph and Schoder Formula

This is another empirical equation that has sometimes been used to design water systems. It is usually written as:

$$h = 0.38\ \frac{v^{1.86}}{D^{1.25}} \quad (18)$$

Where:

h = friction loss, ft. water per 1,000 ft. pipe
v = velocity, ft./sec.
D = internal diameter, ft.

The limitations of the above two empirical equations are obvious and their use cannot be generally recommended. However, where sufficient data and know-how are available they can give satisfactory results.

ELEMENTS OF FIELD PROCESSING—29

How to Calculate Pipe Diameter Needed

by Dr. John M. Campbell

SITUATIONS often occur where it is desired to calculate the line diameter needed for a given flow rate and pressure drop or the capacity of a given size line under fixed pressure conditions. The friction factor f and Reynolds number (Re) both contain diameter and velocity terms. Therefore the use of the correlation of f vs. Re involves a trial-and-error solution if either diameter or capacity is desired.

This trial and error may be prevented by eliminating the variable to be determined. If Q_L represents the volume rate of flow:

$$Q_L = (\pi/4)\ D^2\ v$$

$$Re = \frac{D\,v\,\rho}{\mu} = \frac{Q_L\,\rho}{0.785\,D\,\mu}$$

$$\Delta P_f = \frac{2\,f\,L\,v^2\,\rho}{g\,D} = \frac{32\,f\,L\,Q_L^2\,\rho}{\pi^2\,D^5\,g}$$

where the nomenclature is the same as in preceding parts.

If the equations for f and Re are solved for D you have:

$$D^5 = \frac{32\,f\,L\,Q_L^2\,\rho}{\pi^2\,g\,\Delta P_f} = a\,f \quad (1)$$

and

$$D = \frac{4\,Q\,\rho}{\pi\,\mu\,Re} = \frac{b}{Re} \quad (2)$$

where a and b represent the known terms.

If D is eliminated between Equations 1 and 2:

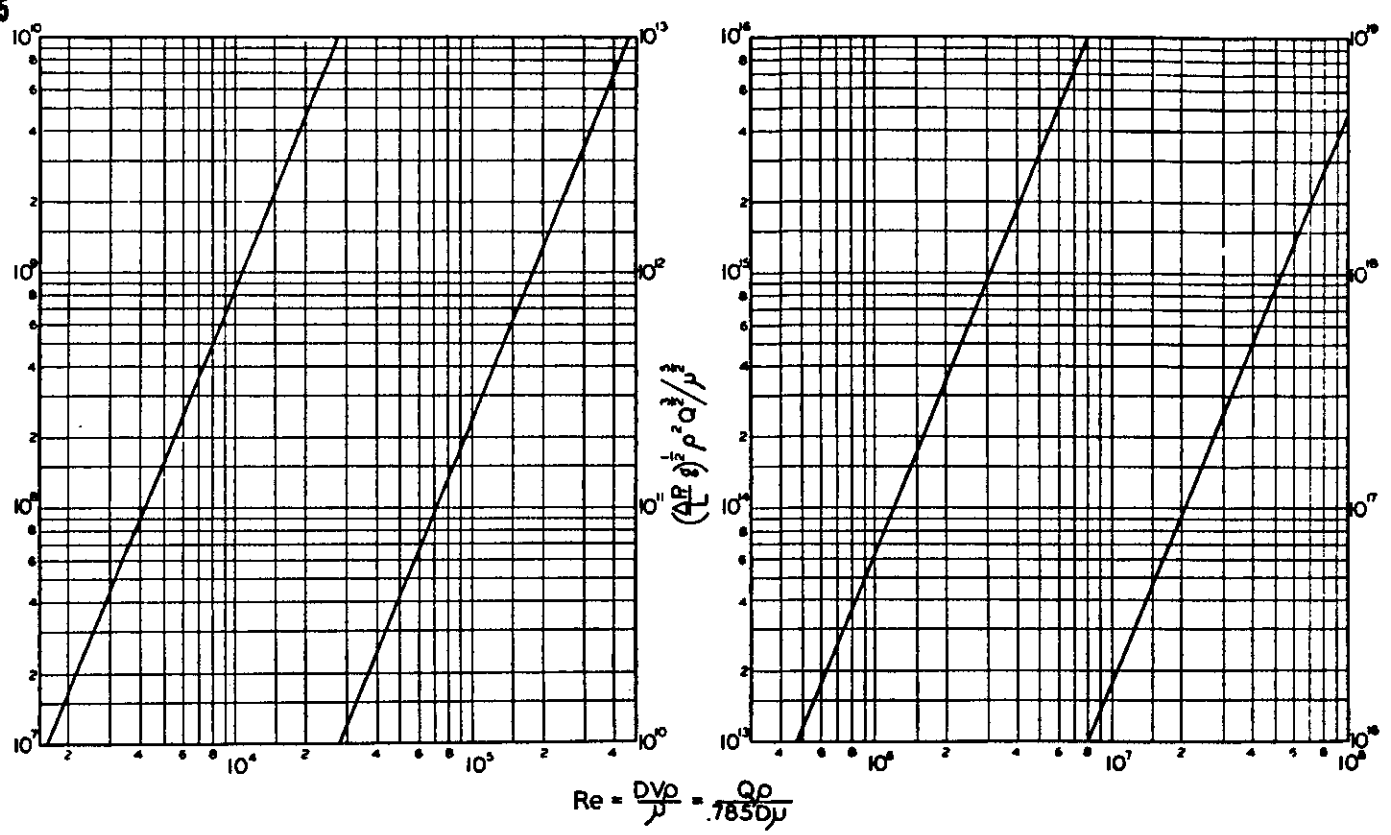
$$b^5/a = f\ (Re)^5 \quad (3)$$

The above expression gives the relationship of f and Re in terms of the quantities necessary to find the diameter D. From a graph of f vs. Re (as shown in Part 28) one may construct a graph of b^5/a as the ordinate and either Re or f as the abscissa. If the value of b and a are substituted back into Equation 3 and the terms combined:

$$\frac{b^5}{a} = \frac{32\,\Delta P_f\,g\,Q_L^5\,\rho^4}{\pi^5\,L\,\mu^5} \quad (4)$$

Fig. 1 is a log-log plot of the square root of the last term of Equation 4, on the ordinate vs. Re on the abscissa. The value of the ordinate may then be calculated knowing the pressure drop, elevations, line length, and liquid properties. This is then extended horizontally to the intersection of the line and then vertically to obtain the value of Re, from which the diameter D may be calculated. By this means the diameter D is obtained, directly.

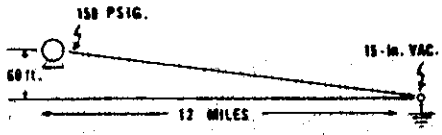
Sensitivity of calculation . . . The correlation shown is derived from the commercial pipe curve in the turbulent



PIPELINE SIZE can be found by this chart. The correlation eliminates trial-and-error estimation of friction factor. Fig. 1.

flow region for the f vs. Re plot in Part 28. Since Re or its equivalent appears in both ordinate and abscissa the calculated value of D is quite insensitive to the friction factor f. This is true not only because of the relationship of the variables but because the inclusion of the same terms on both coordinates tends to give a degree of misleading accuracy. However, Fig. 1 is suitable for all commercial design applications for the error is less than the difference between available pipe sizes.

Example: 12,000 bbl. per day of 1.04 specific gravity brine having an absolute viscosity of 1.12 cp. is pumped 12 miles through Transite pipe to an input well. The pump discharge pressure is 150 psig. and the wellhead pressure will average 15 in. Hg vacuum. The pump discharge is 60 ft. above the wellhead. If Transite pipe is available in diameters of 4, 5, 6, 7, 8. 10, and 12 in. calculate the necessary line size.



Physical properties of brine:

Sp. gr. = 1.04 and $\rho = (1.04)(62.4) = 64.9$ lb./ft.^{3}

Viscosity (abs.) = 1.12 cp.

$\mu = (1.12)(0.000672) = 0.000753$ lb./ft.-sec.

Capacity of line:

$$12{,}000 \text{ bbl./day or } Q_L = \frac{12{,}000 \times 5.61}{24 \times 3{,}600}$$

$$= 0.779 \text{ ft.}^3\text{/sec.}$$

Linear velocity = Q_L/A but D and. therefore, A are unknown.

L = 12 miles or 63,300 ft.

Pressure conditions:

Pump discharge pressure = 150 psig.

Wellhead pressure = 15-in. Hg = 7.35 psi. below atm.

Elevation of wellhead with reference to pump discharge = 60 ft. as shown in the sketch.

Energy balance:

$\Delta P = (150 + 7.35)(144) = 22{,}650$ lb./ft.^{2}

$\Delta X = (60)(64.9) = 3{,}890$ lb./ft.^{2}

So $\Delta P_f = 22{,}650 + 3{,}890 = 26{,}540$ lb./ft.^{2}

(a) Calculate the diameter of the original line:

$$\frac{(\Delta P_f\, g/L)^{1/2}\, \rho^2\, Q_L^{3/2}}{\mu^{3/2}} = \frac{[(26{,}540)(32.2)/63{,}300]^{1/2}\,(0.779)^{3/2}\,(64.9)^2}{(0.000753)^{3/2}}$$

$$= \frac{(3.68)(4{,}200)(0{,}689)}{1.58 \times 10^{-4}} = 6.73 \times 10^{11}$$

From the chart:

$$Re = 1.53 \times 10^5 = \frac{Q\rho}{0.785\, D\, \mu}$$

and

$$D = \frac{(0.779)(64.9)}{(0.785)(1.53 \times 10^5)(0.000753)}$$

$$= 0.559 \text{ ft.}$$

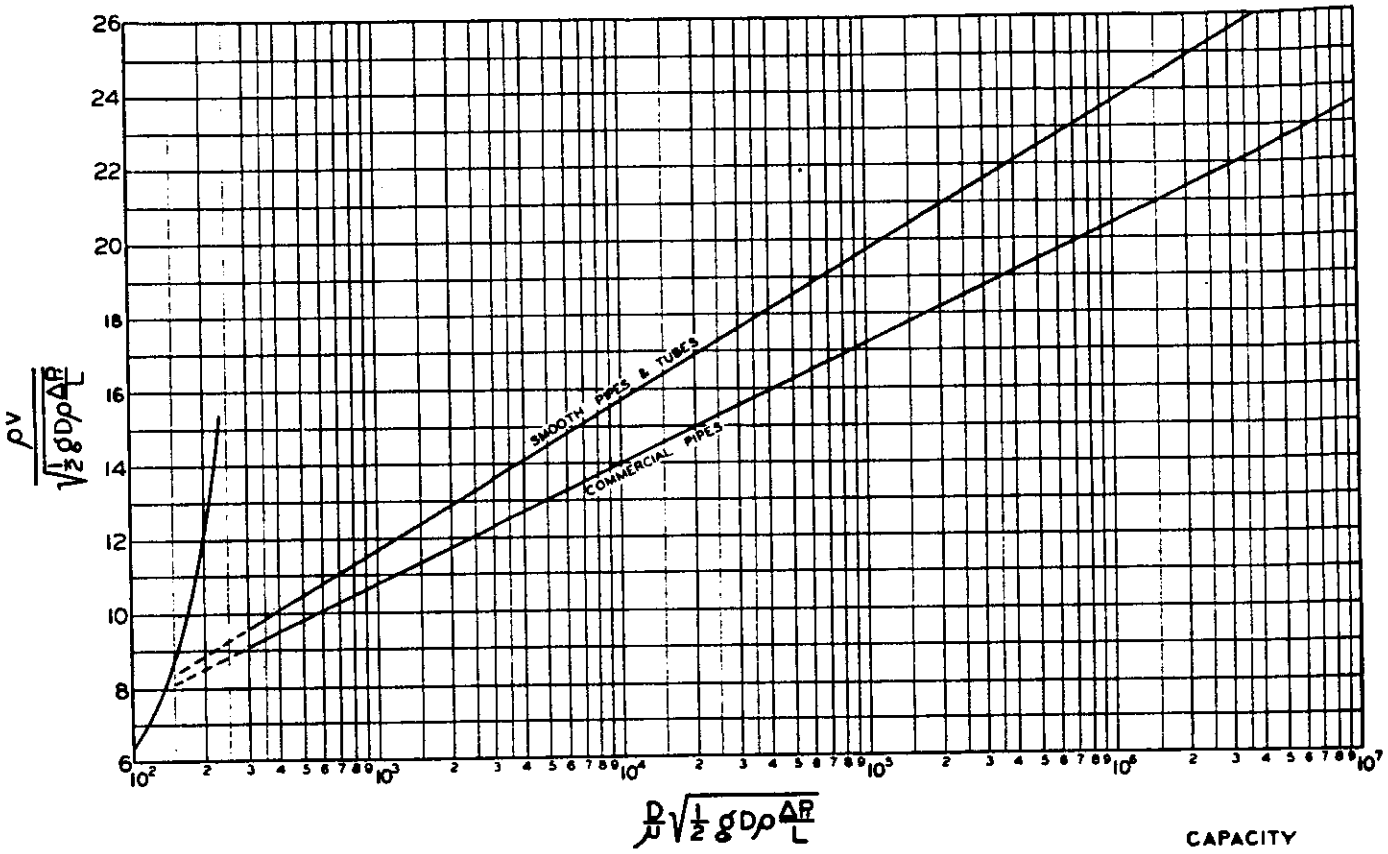
or

$$D = (0.559)(12) = 6.7 \text{ in.}$$

Use 7-in. i.d. Transite pipe.

Nomenclature

D = internal pipe diameter, ft.
f = friction factor
g = gravitational constant, ft./sec.^{2}
L = length, ft.
Q_L = flow rate, ft.^{3}/sec.
ΔP_f = pressure drop due to friction, lb./ft.^{2}
v = velocity, ft./sec.
$\pi = 3.1416$
ρ = density, lb./ft.^{3}
μ = viscosity, lb./ft.-sec.



REPLOT OF FRICTION-FACTOR CURVES permits calculation of capacity of a given size of line when pressure drop and fluid properties are known. Re and f need not be determined. Fig. 1.

ELEMENTS OF FIELD PROCESSING—30

How to Calculate Liquid Line Capacity

by Dr. John M. Campbell

THE calculation of the capacity of a line, with a given pressure drop, is trial and error when using a plot of friction factor vs. Re. However, a direct solution is possible if the capacity is eliminated using the same procedure as that in Part 29. Therefore:

$$Q = \frac{\pi D \mu Re}{4 \rho} = c\,Re \qquad (1)$$

and

$$Q^2 = \frac{\Delta P_f \pi^2 D^5 g}{32 f L \rho} = \frac{d}{f} \qquad (2)$$

when the friction factor and Reynolds number equations are solved for Q. The quantities c and d represent the known values in Equations 1 and 2.

Equating Equations 1 and 2 yields:

$$\frac{d}{c^2} = f\,Re^2 \text{ or } \frac{(d)^{1/2}}{c} = Re\,(f)^{1/2} \qquad (3)$$

If the values of c and d are substituted in Equation 3 and the terms combined:

$$\frac{(d)^{1/2}}{c} = \frac{D}{\mu}[\tfrac{1}{2} D g \rho (\Delta P_f/L)]^{1/2} = Re\,(f)^{1/2} \qquad (4)$$

Fig. 1 has been developed using the relationship shown above and that which was shown between Re and f in Part 28. Fig. 1 is therefore exactly comparable to the f vs. Re correlation.

How to use chart . . . The capacity of a given line is found by solving the quantity on the abscissa, proceeding vertically to the proper line and then horizontally to the ordinate. The latter value may be solved for velocity v from which volume rate of flow Q is obtained. The curves shown for smooth and commercial pipe apply only for the turbulent-flow region and should not be extrapolated into the laminar region.

The approximate equations of the lines are:

For commercial pipe:

$$y = 3.28 \log x + 1 \qquad (5)$$

For smooth pipe and tubes:

$$y = 3.60 \log x + 1.2 \qquad (6)$$

where:

$$y = \frac{\rho v}{[\tfrac{1}{2} g D \rho (\Delta P_f/L)]^{1/2}};$$

$$x = D/\mu\,[\tfrac{1}{2} D g \rho (\Delta P_f/L)]^{1/2}$$

Example: An 8-in. line is converted for crude oil use to handle 20.5° A.P.I. gravity oil with a viscosity of 46.3 cp. If the maximum working pressure is set at 600 psig. on this 18-mile line determine the line capacity if it discharges into atmospheric storage 385 ft. above the pump discharge.

Solution:

$\rho = [141.5/(131.5 + 20.5)]\,(62.4) = (0.93)\,(62.4) =$ **58 lb./ft.³**

$\mu = (46.3)\,(0.000672) = 0.0311$ lb./

ft.-sec.

$L = (18)(5{,}280) = 95{,}000$ ft.

$D = 0.673$ ft.

$\Delta P_f = (600)(144) - (58)(385) =$ **64,000 lb./ft.²**

$$D/\mu\,[\tfrac{1}{2}\,D\,g\,\rho\,(\Delta P_f/L)]^{1/2} = \frac{0.673}{0.0311}\,[\tfrac{1}{2}\,(0.673)\,(32.2)\,(58)\,(64{,}000/95{,}000)]^{1/2}$$

$$= 0.673/0.0311\,(20.5) = \mathbf{4.43 \times 10^2}$$

From Fig. 1:

$\rho v/20.5 = 9.6$ or $v = 3.4$ ft./sec.

$Q = (0.785)\,(0.673)^2\,(3.4) = 1.21$ ft.³/sec. = **18,600 bbl./day.**

The above method produces an answer as accurate as the f vs. Re correlation itself and thus may be used for rigorous pipeline design.

Pipe Roughness

In all of the preceding correlations the pipe-wall conditions have been taken into account only through the use of curves carrying the general designation of commercial and smooth pipe, in the turbulent-flow region. In the laminar-flow region pipe roughness has no significant effect. Through the basic work of Von Karman[1] correlations are available which embody the roughness factor ϵ/D where ϵ is the average depth of the pipe-wall irregularities.

It is then possible to plot f vs. Re with the quantity ϵ/D as parameter to get a family of curves. The apparent accuracy obtained by doing this is largely academic, however, when applied to field lines for even if an average value of ϵ is known for the pipe when new it will change with use as corrosion erosion, scaling, etc., take place. When this is added to the doubtful accuracy of being able to measure an average ϵ for pipe that has been sitting in a pipe yard and the unknown friction losses caused by welding, etc., the roughness correction has less application than might appear at first. The incorporation of roughness into small-plant process-piping design is a sound approach though.

The following are typical of ϵ for various types of pipe:

	ϵ (ft.)
Drawn tubing	0.000005
Steel or wrought iron	0.00015
Galvanized	0.0005
Cast iron	0.00085

Reference

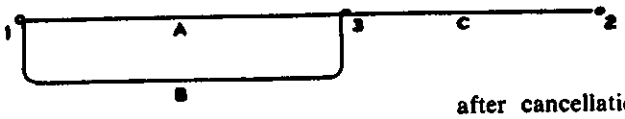
1. Von Karman, T., Nach. Ges. Wiss. Gottingen Fachgruppe, I, 5, 58-76 (1930).

How to Design Loop Systems

by Dr. John M. Campbell

THE need often arises for increasing the flow rate per unit pressure drop in a system. This is normally encountered where it is desired to increase the capacity or where the flow rate is to remain the same with a lower pressure. The former is a common need in expanding systems while the latter usually stems from older systems wherein the pipe or pumping equipment has to be derated for pressure.

The usual and most economical solution to the problem is to place one or more lines parallel to the original, either partially or throughout the entime length. The figure below shows a schematic view of a simple loop system that may be used to illustrate the principles involved.



The loop and the line it parallels are usually equal in length, for all practical purposes, although conditions sometimes dictate a loop of different length. If the lines are of equal length, however,

$$(\Delta P_f/L)_A = (\Delta P_f/L)_B \text{ and}$$

$$(\Delta P/L)_A = (\Delta P/L)_B \qquad (1)$$

since the lines have two points in common and the change in elevation between Points 1 and 3 is the same. The friction drop in the uplooped section plus that in the looped section must also equal the total friction drop.

The total flow must be divided between lines A and B in some ratio so therefore:

$$Q_A + Q_B = Q_C \qquad (2)$$

Equations 1 and 2 outline the simple relationships existing between the lines so it is necessary to apply these together with the basic equations previously discussed. Several approaches to the problem may be taken.

Use of flow equation . . . It has been developed that the friction factor f is related to Reynolds number (Re) by the equation:

$$f = K/Re^n \qquad (3)$$

where K and n are constants that depend on the type of flow, and that:

$$\Delta P_f = \frac{2 f L v^2 \rho}{g D} \qquad (4)$$

If Equation 3 is substituted into Equation 4 the resultant expression may in turn be incorporated into Equation 1 yielding:

$$\frac{L_A v_A^2}{D_A (D_A v_A)^n} = \frac{L_B v_B^2}{D_B (D_B v_B)^n} \qquad (5)$$

after cancellation of like terms, or:

$$v_A/v_B = [L_B/L_A (D_A/D_H)^{1-n}]^{1/2-n} = C \qquad (6)$$

When $L_A = L_B$:

$$v_A/v_B = (D_A/D_H)^{1-n/2-n} = C \qquad (7)$$

The total flow rate Q_C from Equation 2 therefore equals:

$$0.785\ D_A^2 v_A + 0.785\ D_B^2 v_B = Q_C \qquad (8)$$

or

$$0.785\ D_A^2 C v_B + 0.785\ D_B^2 v_B = Q_C \qquad (9)$$

Equation 9 then enables one to calculate the velocity in both looped lines once the ratio of velocities C is obtained from either Equation 6 or 7. For the usual case of turbulent flow, K = 0.04 and n = 0.172 (see Part 28).

Use of chart . . . The same result may be obtained by using the correlation between $D/\mu\,[\frac{1}{2} D g \rho (\Delta P_f/L)]^{1/2}$ and $\rho v/[\frac{1}{2} D g \rho (\Delta P_f/L)]^{1/2}$ presented in Part 30. Any value of ΔP_f may be assumed and the value calculated for the first expression, and placed on the abscissa for each parallel line in the loop system. The corresponding value found on the ordinate enables one to calculate a value of v for each line.

The value of velocity thus found is not the right one unless by mere coincidence the correct value of ΔP_f was assumed (which is unlikely) but the relative values obtained may be used to obtain the percentage of flow through each line:

$$Q_A/Q_C\ (100) = \frac{v_A D_A^2}{v_A D_A^2 + v_B D_B^2}\ (100) \qquad (10)$$

The applicable percentage times the total flow rate then yields the actual flow rate through each line.

After the flow rate in either parallel line has been determined by one of the above methods the relationship between f and Re (Part 28) may be used to find the respective ΔP_f terms needed to solve the problem for:

$$(\Delta P_f/L)_A X_A + (\Delta P_f/L)_C (L - X) = \text{total } \Delta P_f \qquad (11)$$

Example: A water-flood operator has developed his property so that the present 6-in. raw-water line, which is exactly 8 miles long, delivers 15,000 bbl. per day of fresh water under a gravity flow head of 610 ft. of water.

A new development program for adjacent acreage requires that the capacity of the water-treating plant and the line be increased to 28,000 bbl. per day. If 8-in. pipe is available determine how much must be looped into the original line.

Solution:

$$Q = \frac{(28{,}000)\ (5.61)}{(24)\ (3{,}600)}$$

$$= 1.82 \text{ cu. ft./sec.}$$

$$v = \frac{1.82}{0.785\ (0.506)^2} = \frac{1.82}{0.201}$$

$$= 9.05 \text{ ft./sec.}$$

Physical properties of water:
$\rho = 62.4$ lb./ft.^{3}
$\mu = 1 \times 0.000672$ lb./ft.-sec.

When handling 28,000 bbl./day, ΔP_f/mile, for 6-in. line:

$$Re = \frac{(0.506)(9.05)(62.4)}{0.000672} = 4.25$$

$\times 10^5$ and f = 0.0042

ΔP_f/mile

$$= \frac{(2)(0.0042)(5{,}280)(62.4)(9.05)^2}{(32.2)(0.506)}$$

= 13,900 lb./ft.^{2}/mile

For loop section, assume ΔP_f/1,000 ft. = 10,000 lb./ft.^{2}

For 6-in. pipe solving for capacity:

$$\frac{0.506}{0.000672}[(16.1)(0.506)(62.4)(10)]^{1/2}$$

$$= \frac{(0.506)(71.3)}{0.000672} = 53{,}750$$

and from chart: $\rho v/71.3 = 16.4$:

$$v = \frac{(16.4)(71.3)}{62.4} = 18.73 \text{ ft./sec.}$$

and Q = (18.73) (0.201)

= 3.77 cu. ft./sec.

For 8-in. pipe, solving for capacity:

$$\frac{(0.673)}{0.000672}[(16.1)(0.673)(62.4)(10)]^{1/2}$$

$$= \frac{(0.673)(82.2)}{0.000672} = 82{,}400$$

and from chart: $\rho v/82.2 = 16.9$:

$$v = \frac{(16.9)(82.2)}{62.4} = 22.3 \text{ ft./sec.}$$

and Q = (0.355) (27.3)

= 7.91 cu. ft./sec.

$$Q_{8"} = \frac{3.77}{3.77 + 7.91}(100)$$

= 32.3 per cent

Therefore:

$Q_{8"} = 67.7$ per cent

Applying volume per cent to the daily quantity of water to be handled:

6-in. = 0.323 × 28,000

= 9,050 bbl./day

8-in. = 0.677 × 28,0000

= 18, 950 bbl./day

or, on the basis of 1.82 cu. ft./sec.

$Q_{6"} = 0.323 \times 1.82$

= 0.588 cu. ft./sec.

$Q_{8"} = 0.677 \times 1.82$

= 1.232 cu. ft./sec.

To find ΔP_f/mile of loop section:

Taking 6-in. line:

$$v = \frac{0.588}{0.785(0.506)^2} = \frac{0.588}{0.201}$$

= 2.925 ft./sec.

$$Re = \frac{(0.506)(2.94)(62.4)}{0.000672} = 1.38$$

$\times 10^5$ and f = 0.005

$$\Delta P_f = \frac{(2)(0.005)(5{,}280)(62.4)(2.94)^2}{(32.2)(0.506)}$$

= 1,730 lb./ft.^{2}

Checking, take 8-in. line:

$$v = \frac{1.229}{0.785(0.673)^2} = \frac{1.229}{0.355}$$

= 3.47 ft./sec.

$$Re = \frac{(0.673)(3.47)(62.4)}{0.000672} = 2.17$$

$\times 10^5$ and f = 0.0047

$$\Delta P_f = \frac{(2)(0.0047)(5{,}280)(62.4)(3.47)^2}{(32.2)(0.673)}$$

= 1,720 lb./ft.^{2}

The available ΔP_f on the line:

(610) (62.4) = 38,000 lb./ft.^{2}

So, if X = miles of loop and (8 − X) = miles of unlooped 6-in.:

(X) (1.730) + (8 − X) (13,900)

= 38,000

1,730 X + 111,000 − 13,900 X

= 38,000

12,160 X = 73,000

X = 6.0 miles of loop line.

The above basic procedure may be used for a loop system containing any number of parallel lines. It may also be shown that the position of the loop along the line makes no difference in the total capacity.

Complex Liquid Gathering Systems

by Dr. John M. Campbell

THE situation often arises when it is convenient to use several diameters of pipe in series. Fluid principles may also be applied in this case after proper analysis of the conditions present.



In the above sketch L and D represent the length and diameter respectively of two different size lines joined in series. Therefore:

$$Q_{LA} + Q_{LB} = Q_{L\ Total}$$

and

$$\Delta P_{fB} + \Delta P_{fA} = \Delta P_{f\ Total}$$

However, no one equation may be written between points (1) and (2) because of the varying diameter. It should be possible to write an equation for section 1-3 and one for 3-2, solve each for Q_L, and permit one to solve for the remaining unknown P_3.

A more flexible and convenient solution is afforded by determining a single-diameter line that is equivalent to the system shown. The equation

$$\Delta P_f = \frac{32\, f\, L\, Q_L^2\, \rho}{\pi^2\, g\, D^5}$$

may be written for one section. Then the equation is written for a hypothetical section having the same diameter as the other section but of such length that the friction drop is the same as that first written. Then:

$$\Delta P_f' = \frac{CL'}{D_A^5} = \Delta P_{fB} = C\frac{L_B}{D_B^5} \quad (1)$$

where L' is defined as the length of line of diameter D_A that will give the same friction loss as L_B feet of line D_B.* If the constants are canceled and Equation 1 rearranged:

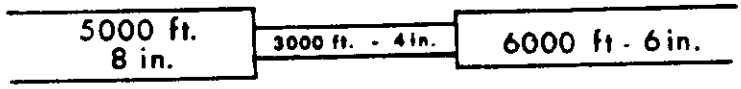
$$L = L_B\,(D_A/D_B)^5 \quad (2)$$

If D' is defined as the diameter of line of length L_A that will give the same friction loss as L_B feet of line D_B, Equation 1 may be rearranged to yield:

$$D' = D_B\,(L_A/L_B)^{1/5} \quad (3)$$

Equation 2 is particularly useful, for the system shown may then be regarded as consisting of $L' + L_A$ feet of line of diameter D_A. Once this equivalence is established the problem may be solved in the conventional manner.

Example . . . Express the equivalent length of the following system in terms of 4-in. pipe, assuming that the nominal diameter is equal to the actual inside diameter.



Solution . . . Expressing the diameter in terms of 4-in. pipe is arbitrary for it could also be expressed in terms of the 8-in. or 6-in. From Equation 3:

$$L'_{4\text{-}} = (5{,}000)\,(4/8)^5 + (6{,}000)\,(4/6)^5$$

$$= 156 + 790 = 946 \text{ ft.}$$

$$\text{Total length} = L'_{4\text{-}} + L_{4\text{-}} = 946 + 3{,}000 = 3{,}946 \text{ ft.}$$

The system shown may therefore be considered as equivalent to one consisting of 3,946 ft. of 4-in. pipe. Capacity or pressure-drop calculations may then be made on the latter system using the correlations previously presented.

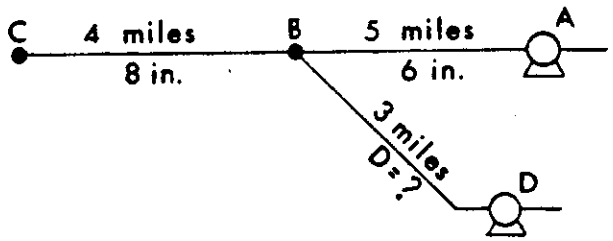
Complex Systems

Most gathering systems contain one or more branch lines. On a system of this type no single "recipe" is possible because of the innumerable variations possible but application of the principles presented (Parts 27-30) will enable one to calculate the necessary information, provided minimum needed data are available.

The problem below is a simple case that illustrates the general type of problem that often occurs.

Example . . . A gathering line, 9 miles long, has been laid to handle 25,000 bbl. per day of 35° A.P.I. crude oil (17.16 cp.) and is made up of 5 miles of 6-in. pipe, on the upstream end and 4 miles of 8-in. pipe. Later it is decided to tie in another line at the junction of the 6-in. pipe and 8-in. pipe, the new line being 3 miles long and shall handle 10,000 bbl. per day of the same crude oil from an extension of the same field.

Calculate the size of the new line with the limitation that the maximum working pressure of the new pump shall not exceed the maximum working pressure of the pump at the head of the 6-in. line. What are the maximum theoretical horsepower requirements for both pumps assuming atmospheric suction?



Solution . . . In effect the problem states that ΔP_{DB} is to equal ΔP_{AB}. Therefore if ΔP_{AB} is calculated it may in turn be used for line DB. The pressure drop in BC is common to both and may be added to determine the total pressure at points C and D.

Considering 8-in. line (BC): Capacity = 35,000 bbl./day of 35° A.P.I. crude.

μ = 17.16 cp. or 0.01153 lb./ft.-sec.

*The C term is not strictly constant for lines of different diameters since it includes the variable f. The error in assuming C constant is small provided the difference in pipe diameters is not large. For highest precision, the diameter exponent of 4.828 should be used rather than 5.

$\rho = (0.85)\ (62.4) = 53$ lb./ft.^{3}

$D = 8.07/12 = 0.673$ ft.

$$v = \frac{(35{,}000)\ (5.61)}{(24)\ (3{,}600)\ (0.785)\ (0.673)^2}$$

$= 6.4$ ft./sec.

$L = (4)\ (5{,}280) = 21{,}120$ ft.

$$Re = \frac{(0.673)\ (6.4)\ (53)}{0.01153} = 19{,}750$$

and $f = 0.0075$

$$\Delta P_{BC} = \frac{(2)\ (0.0075)\ (21{,}120)\ (53)\ (6.4)^2}{(32.2)\ (0.673)}$$

$= 31{,}700$ lb./ft.^{2} or 220 lb./in.^{2}

Considering 6-in. line (AB): Capacity = 25,000 bbl. per day—same crude.

$D = 0.506$ ft.

$$v = \frac{(25{,}000)\ (5.61)}{(24)\ (3{,}600)\ (0.785)\ (0.506)^2}$$

$= 8.08$ ft./sec.

$L = (5)\ (5{,}280) = 26{,}400$ ft.

$$Re = \frac{(0.506)\ (8.08)\ (53)}{0.01153} = 18{,}760$$

and $f = 0.0076$

$$\Delta P_{AB} = \frac{(2)\ (0.0076)\ (26{,}400)\ (53)\ (8.08)^2}{(32.2)\ (0.506)} = 85{,}100 \text{ lb./ft.}^2 \text{ or } 591 \text{ lb./in.}^2$$

$\Delta P = 31{,}700 + 85{,}100$

$= 116{,}800$ lb./ft.^{2} or 810 lb./in.^{2}

According to stated problem, pump at D is to put up same ΔP as pump at A, so:

$\Delta P_{BD} = 85{,}100$ lb./ft.^{2}

$L = (3)\ (5{,}280) = 15{,}840$ ft.

$$Q = \frac{(10{,}000)\ (5.61)}{(24)\ (3{,}600)} = 0.65 \text{ ft.}^3\text{/sec.}$$

Using the chart for pipeline size in Part 29:

$$\frac{(\Delta P_f\, g/L)^{1.2}\, \rho^2\, Q^{3/2}}{\mu^{3/2}} = \frac{[(85{,}100)\ (32.2)/15{,}840]^{1/2}\ (52)^2\ (0.65)^{3/2}}{(0.01153)^{3/2}}$$

$$= \frac{(13.4)\ (2{,}800)\ 0.524)}{0.0000143} = 1.37 \times 10^9$$

Then:

$$Re = 12{,}000 = \frac{Q\rho}{0.785\, D\mu}$$

$$= \frac{(0.65)\ (53)}{(0.785)\ (D)\ (0.01153)}$$

$$D = \frac{(0.65)\ (53)}{(0.785)\ (12{,}000)\ (0.01153)}$$

$= 0.317$ or 3.8 in. Take 4-in. pipe.

ΔP using 4-in. pipe:

$D = 4.03/12 = 0.336$ ft.

$$v = \frac{(10{,}000)\ (5.61)}{(24)\ (3{,}600)\ (0.785)\ (0.336)^2}$$

$= 7.32$ ft./sec.

$$Re = \frac{(0.336)\ (7.32)\ (53)}{0.01153} = 11{,}300$$

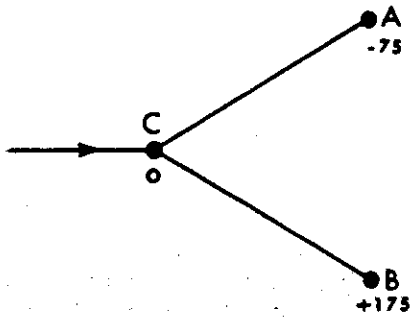
and $f = 0.0086$

$$\Delta P_{DB} = \frac{(2)\ (0.0086)\ (15.840)\ (53)\ (7.32)^2}{(32.2)\ (0.336)} = 71{,}500 \text{ lb./ft.}^2 \text{ or } 796 \text{ lb./in.}^2$$

$$Hp._D = \frac{(0.65)\ (103{,}200)}{550} = 122 \text{ hp.}$$

$$Hp._A = \frac{(1.624)\ (116{,}800)}{550} = 345 \text{ hp.}$$

Split flow . . . A common problem also encountered is where the flow from a single line is split into two or more parts as in the figure below.



In the rare case where the pressures and elevations at A and B are equal this may be handled as a loop problem—in all others the calculation of flow is theoretically a trial and error process. However, this tedious approach may usually be eliminated by the following method:

1. Assume a pressure at point C that is high (greater than 100,000 lb./ft.^{2}).
2. Calculate ΔP_f for lines CA and CB.
3. Using the correlation:

$$(D/\mu)\ [\tfrac{1}{2}\ D\, g\, \rho\, (\Delta P_f/L)]^{1/2}$$

vs. $\rho v/[\tfrac{1}{2}\ D\, g\, \rho\, (\Delta P_f/L)]^{1/2}$

(Part 30) calculate Q_L for both lines.

4. Knowing the total flow rate at C and the relative Q_L's from (3) above, calculate the actual flow rate for each line.
5. Using the actual flow rates from (4) above calculate ΔP_f for lines CA and CB using the correlation of f vs. Re (Part 28).
6. Calculate pressure at C for both lines.
7. If the two pressures at C check within a few per cent the actual flow rates calculated in (4) above may be used. If not, a new value must be assumed for the pressure at C and the process repeated.

In most instances the above process is only necessary one time, particularly if the pressure at C is assumed high enough. It will often work even if the assumed pressure and that calculated in Step 6 differ widely. As a rule of thumb the assumed pressure should be about 15 times the maximum elevation change, expressed in pounds per square foot.

TABLE 1—CALCULATION OF PSEUDOCRITICAL TEMPERATURE AND PRESSURE

Component—	Mole Fr., N	P_c	T_c	NP_c	NT_c
Methane	0.82	673	344	551	282
Ethane	0.10	708	549	71	55
Propane	0.05	617	666	32	33
Isobutane	0.01	530	732	5	8
n-Butane	0.02	551	766	11	15
	1.00			$P_c = 670$	$T_c = 393$

TABLE 2—SUMMARY OF PROPERTIES OF VARIOUS COMMON GASES

Gas—	T_c-°R.	P_c-psia.	Sp. gr. (air=1.0)	Mol. wt.
Methane	344	673	0.55	16
Ethane	549	708	1.04	30
Propane	666	617	1.52	44
i-Butane	732	530	2.01	58
n-Butane	766	551	2.01	58
i-Pentane	829	482	2.49	72
n-Pentane	846	485	2.49	72
Hexane	914	434	2.98	86
Heptane	972	397	3.46	100
Octane	1,024	361	3.94	114

tion in gas volume is greater than that due to external pressure alone. However, at high pressure the fraction of the total volume occupied by the molecules becomes high, making the first two terms in the bracketed portion the controlling factors. Consequently, a curve of Z versus pressure at constant temperature should initially decrease, reach a minimum, and then increase.

The value of a and b may be related to the critical constants, i.e., pressure, temperature, and volume. Any gas may be liquefied by an increase in pressure provided that the temperature is at or below a value known as the absolute critical temperature T_c. The pressure needed for liquefaction at T_c is the critical pressure P_c, the volume being V_c. Each of the values T_c, P_c, and V_c are fixed characteristics of a pure, single component gas such as methane. Values of these and other constants are shown in Table 1.

The equations for a and b are

$$b = V_c/3 = RT_c/8P_c;$$
$$a = P_cV_c^2 = (27/64)\,R^2T_c^2/P_c \quad (6)$$

which are obtained by writing the van der Waals equation at critical conditions, extracting the three roots, and expanding the terms using the binomial theorem.

Law of Corresponding States

If the values of P, V, and T in van der Waals' equation are expressed as fractions of the corresponding critical states

$$P_r = P/P_c,\ V_r = V/V_c$$
$$\text{and } T_r = T/T_c \quad (7)$$

Further, if a and b are expressed in terms of the critical constants an equation

$$(P_r + 3/V_r)\,(3V_r - 1) = 8T_r \quad (n=1)\ (8)$$

results which is known as the reduced equation of state. It can be seen that Equation 8 contains no terms that are characteristic of a given substance. Therefore, it should apply to all substances and leads to the definition of what is known as the Law of Corresponding States.

This law states that any two or more substances at the same reduced pressure P_r and reduced temperature T_r are in the same state and a universal equation could be written for all substances in both the liquid and vapor states using these coordinates. This concept is not entirely rigorous but yields results sufficiently accurate for most routine engineering calculations.

Pseudocritical Conditions

Inasmuch as most gases are mixtures it becomes necessary to determine the critical constant of the mixture. This may be done experimentally, but the procedure is tedious, costly, and subject to error. Thus, it is customary to calculate pseudocritical constants which are suitable for use in most engineering calculations.

The appropriate pseudocritical constant is obtained by multiplying the corresponding critical value of a pure component by the mole fraction of that component in the mixture, as shown in Table 1.

The value of the critical constants determined in this manner are only approximate but have sufficient accuracy for most engineering calculations. The same scheme may also be used for calculating average molecular weight, specific gravity, etc.

Fig. 1 is commonly used correlation for finding critical pressure and temperature developed by Matthews, Roland, and Katz[1] from representative data. It is reasonably accurate for all applications, in the absence of a gas composition, where the specific gravity is known.

Reference

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ELEMENTS OF FIELD PROCESSING—34

Calculation of the Compressibility Factor Z

By Dr. John M. Campbell

THE compressibility factor Z offers an accurate and yet convenient method of determining the P-V-T properties of an actual gas according to the equation:

$$PV = ZnRT \quad (1)$$

The Law of Corresponding States (Part 33), although not an exact one, further offers a satisfactory method of preparing a general correlation. Such a correlation of course is not entirely rigorous for it will vary slightly with the gas, but the error is substantially less than the error present in the engineering calculations in which the resultant value of Z is incorporated.

Figs. 1 and 1a are correlations for finding Z from the work of Nelson and Obert[1] and are based on data from 30 common gases, including those components present in natural gas. Consequently they are applicable to most gas P-V-T problems.

Example: Determine the value of Z for 0.6 specific gravity gas (T_c = 360° R., P_c = 672 psia.) at 80° F. and 500 psia.

Solution:

$$P_r = \frac{500}{672} = 0.744$$

$$\text{and } T_r = \frac{80 + 460}{360} = 1.5$$

From Fig. 1, Z = 0.935 by interpolation.

Reference

1. Nelson, L. C., and Obert, E. F., Chem. Eng., pp. 203-08, July (1954).

Natural-Gas P-V-T Relationships

By Dr. John M. Campbell

A METHOD of predicting the pressure-volume-temperature (P-V-T) relationships of natural gas is essential in all phases of field processing.

The basic relationship is one derived from the Boyles and Charles law for an ideal gas.

$$PV \text{ is proportional to } nT \quad (1)$$

or

$$PV = nRT \quad (2)$$

The value R is a proportionality constant that has become known as a "gas constant," through usage, values of which are available throughout the literature.

The concept of an ideal gas is a convenient one but involves the assumption that the gas molecules have neither volume nor mutual attraction. Consequently, at elevated pressures (particularly above 50 psi.), it is necessary to have other relationships, known as equations of state. Several of these equations of state, including the Beattie-Bridgman, Keyes, etc., have a number of arbitrary constants that make their use tedious and limit their application to specialized uses.

NOMENCLATURE

Term	Meaning	Usual Units
P	= absolute pressure	lb./ft.² abs. or psia.
P_c	= absolute critical pressure	lb./ft.² abs. or psia.
P_r	= reduced pressure, P/P_c	dimensionless
T	= absolute temperature	°R. or °K.
T_c	= absolute critical temperature	°R. or °K.
T_r	= reduced temperature, T/T_c	dimensionless
V	= volume or specific volume	cu. ft. or cu. ft./lb.
V_c	= critical volume	cu. ft. or cu. ft./lb.
V_r	= reduced volume, V/V_r	dimensionless
n	= moles of gas present	lb./mole
R	= gas constant	1,544 (ft.-lb.)/(lb.-mole-°R.) or 10.78 (ft.³-lb.)/(in.²-lb.-mole-°R.)
a	= van der Waals' constant	units used for P, V, and T
b	= van der Waals' constant	units used for P, V, and T
Z	= compressibility factor	dimensionless

NOTE: All equations used are dimensionally homogeneous, so that consistent units must be used throughout.

...The van der Waals equation is the first proposed that has received widespread usage.

$$(P + n^2a^2/V^2)(V - nb) = nRT \quad (3)$$

The values a and b are arbitrary constants.

The principal use of this equation comes from its ability to qualitatively predict the nature of the deviation of real from ideal gases, for quantitatively it does not yield results of satisfactory accuracy. In effect, this equation of state serves as the basis for the concept of compressibility, which is expressed through the factor Z. This factor is incorporated into Equation 2 to give

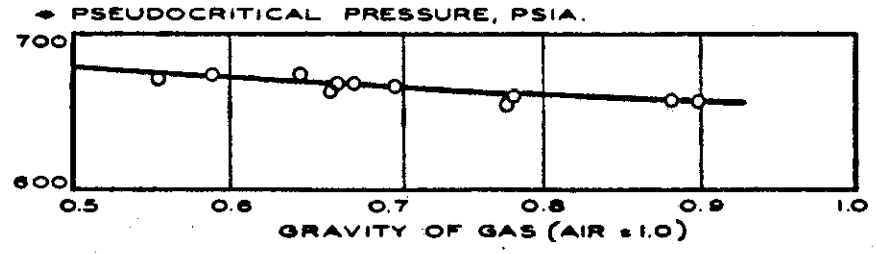
$$PV = ZnRT \quad (4)$$

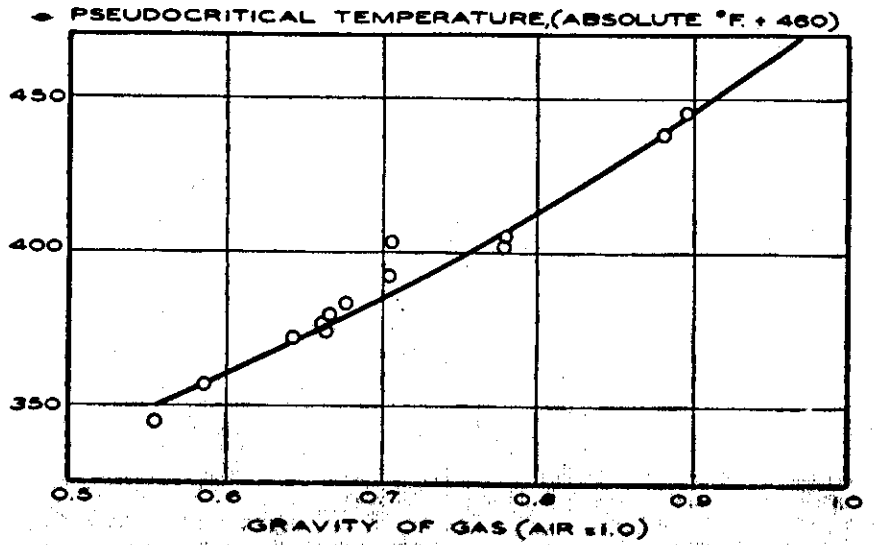
where Z is the compressibility, or super compressibility, factor. If the left side of Equation 3 is expanded, for n = 1, and transposed, the compressibility factor for a van der Waals gas is

$$Z = 1 + (1/RT) \times [bP + (ab/V^2) - (a/V)] \quad (5)$$

The quantity in brackets has a value of zero when a and b equal zero, which makes Z = 1, a criterion of an ideal gas. Consequently, the constants a and b are a measure of the deviation of an actual gas from the ideal state, where a corrects for molecular forces and b for the molecular volume.

If the pressure is isothermally increased on an actual gas, Equation 5 furnishes a means of predicting the expected variation of Z. At low pressures molecular force, rather than volume, is controlling and the reduc-





CRITICAL PRESSURES and temperatures for a gas can be estimated by the above curves based on gas gravity. Fig. 1.

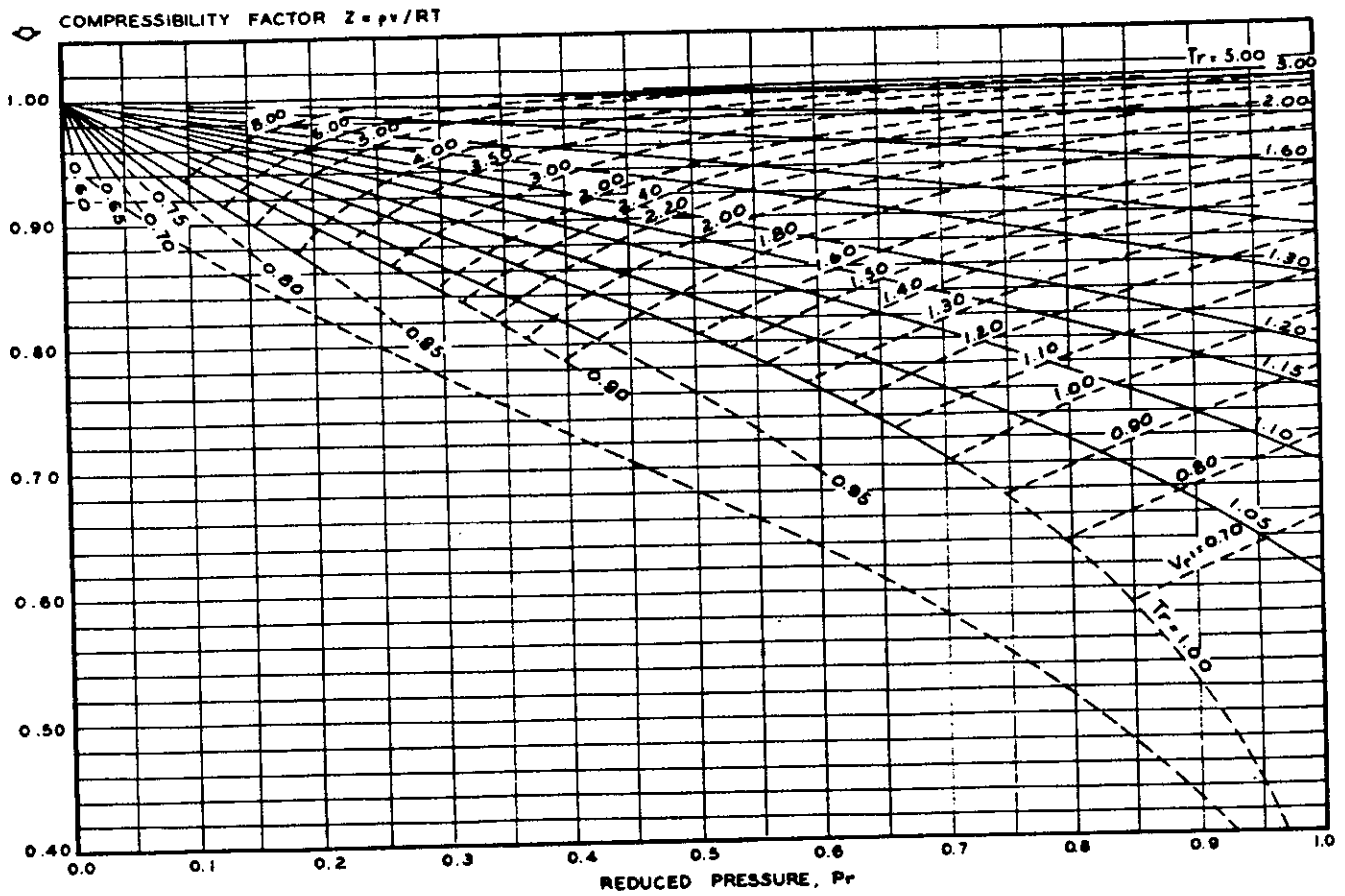


Fig. 1—Compressibility factors in the low pressure range.

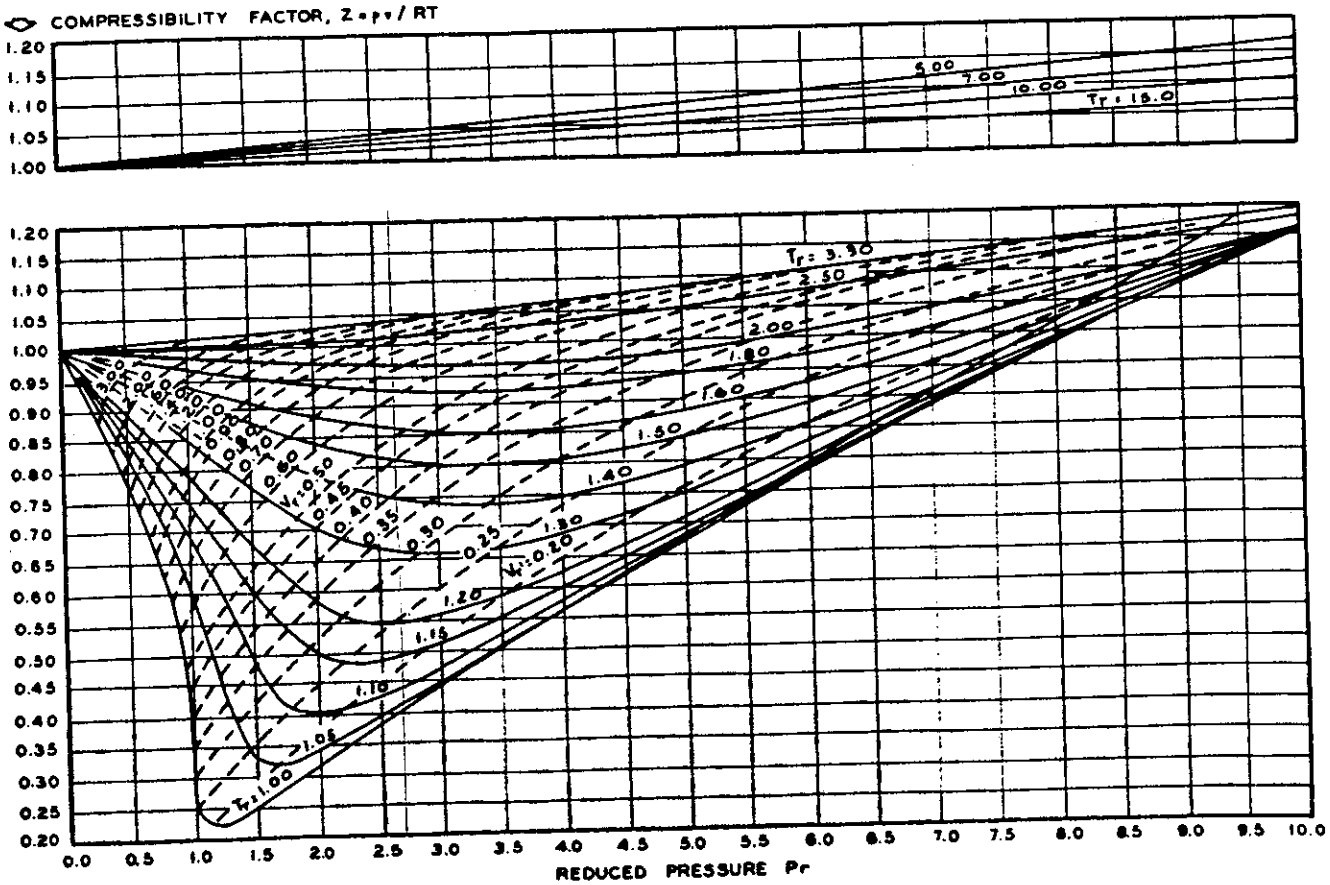


Fig. 1a—Compressibility factors for high pressures and high temperatures.

Principles of Gas Flow

By Dr. John M. Campbell

THE FUNDAMENTAL thermodynamic equations used for liquid flow may also be used for gases. A logical starting point is the equation developed in Part 27, modified to read:

$$-\int PdV + \Delta(PV) + \Delta X + \frac{v_2^2 - v_1^2}{2g} = -W_f - W \quad (1)$$

where:

- V = volume or specific volume
- P = absolute pressure
- X = elevation above an arbitrary plane
- v = velocity
- g = gravitational constant
- W = work done by system
- W_f = total of irreversible effects in system—usually attributed to friction

In order to conveniently handle Equation 1 several assumptions are often made:

1. That no external work is done by or on the system, i.e., W = 0.

2. That natural gas behaves as an ideal gas, i.e., $P_1V_1 = P_2V_2$.

3. That the flow is isothermal.

4. That the changes in elevation on a long pipeline are negligible, i.e., $\Delta X = 0$.

All of the above assumptions except that of ideal gases are usually satisfactory on the average pipeline. However, those equations derived on this basis usually contain a compressibility-factor correction if designed to be used at pressures over several hundred pounds per square inch (see Parts 34 and 36). The assumption of isothermal flow has little effect on the final accuracy, which may be proven by assuming adiabatic conditions, the opposite extreme.

When applying the above assumptions to Equation 1 and rearranging:

$$W_f = \int PdV - \frac{v_2^2 - v_1^2}{2g} \quad (2)$$

Differentiating:

$$dW_f = PdV - \frac{vdv}{g} \quad (3)$$

Equation 3 is as far as one may go with thermodynamics alone inasmuch as it does not provide a means of solving for the loss in energy due to friction. Experiments show that the work done in overcoming friction through distance dL is proportional to the surface in contact with the fluid, approximately proportional to the square of velocity and proportional to the density of the fluid. By including a proportionality constant "f" the equation may be written as:

$$\text{Frictional resistance} = (f)\ (dL)\ (\pi D)\ (v^2/2g)\ (\rho) \quad (4)$$

or since work equal force times distance:

$$\text{Work} = (f)\ (dL)\ (\pi D)\ (v^2/2g)\ (\rho)\ (dL) \quad (5)$$

If Equation 5 is divided by the weight of fluid in the length dL ($A \rho\, dL$) and the result applied to circular conduit:

$$dW_f = \frac{2f\,v^2\,dL}{gD} \quad (6)$$

and Equation 3 reduces to:

$$\frac{2f\,v^2\,dL}{gD} = PdV - \frac{vdv}{g} \quad (7)$$

By appropriate algebra, application of the ideal gas law and subsequent integration:

$$N = A\left[\frac{gM\,(P_1^2 - P_2^2)/2RT}{2fL/D + \ln P_1/P_2}\right]^{1/2} \quad (8)$$

TABLE 1—SUMMARY OF GAS-FLOW EQUATIONS BASED ON EQUATION 10

FRICTION FACTOR	EQUATION	NAME
EQUATIONS WITH CONSTANT "f"		
CONSTANT	$Q_h = 43.87\, D^{5/2} \left[\frac{P_1^2 - P_2^2}{L}\right]^{1/2}$	COX [2]
CONSTANT	$Q_h = 48.61\, D^{5/2} \left[\frac{P_1^2 - P_2^2}{L}\right]^{1/2}$	RIN [3]
CONSTANT	$Q_h = 50.03\, D^{5/2} \left[\frac{P_1^2 - P_2^2}{L}\right]^{1/2}$	TOWL [4]
CONSTANT	$Q_h = 48.47\, D^{5/2} \left[\frac{P_1^2 - P_2^2}{L}\right]^{1/2}$	PITTSBURGH [4]
EQUATIONS WHERE f IS SOME FUNCTION OF DIAMETER		
DIAMETER FUNCTION	$Q_h = 42 \left[D^{5/2} + \frac{D^3}{30}\right] \left[\frac{P_1^2 - P_2^2}{L}\right]^{1/2}$	OLIPHANT [5]
$f = 0.0044\left[1 + \frac{12}{7D}\right]$	$Q_h = 3.30 \left[\frac{D^5}{0.0044\left(1 + \frac{12}{7D}\right)}\right] \left[\frac{P_1^2 - P_2^2}{L}\right]^{1/2}$	UNWIN [6]
$f = \frac{0.008}{D^{1/3}}$	$Q_h = 36.83\, D^{8/3} \left[\frac{P_1^2 - P_2^2}{L}\right]^{1/2}$	WEYMOUTH [7]
DIAMETER FUNCTION	$Q_h = 50.93 \left[\frac{D^5}{1 + \frac{3.6}{D} + 0.03D}\right]^{1/2} \left[\frac{P_1^2 - P_2^2}{L}\right]^{1/2}$	SPITZGLASS [8]
$\frac{1}{f} = \left(\frac{QG}{D}\right)^{0.1461}$	$Q_h = 435.87 \left(\frac{T_S}{P_S}\right)^{1.0788} D^{2.6182} \left[\frac{P_1^2 - P_2^2}{L T_f G_L}\right]^{0.5394}$	PANHANDLE [9]
$\left(\frac{1}{f}\right)^{1/2} = 16.7\, \frac{QG\ 0.01961}{D}$	$Q_h = 737 \left(\frac{T_S}{P_S}\right)^{1.02} \left[\frac{P_1^2 - P_2^2}{0.961 L T_f}\right]^{0.510} D^{2.530}$	MODIFIED PANHANDLE [10]
FORMULAS WHERE f IS FUNCTION OF REYNOLDS MODULUS		
VALUE OF f IN EQUATION 10		NAME
$f = 0.0048 \left[\frac{3600 \times 53.33 T_S}{144\, Q_h G P_S}\right]^{1/7}$		FRITZSCHE [11]
$f = 0.0018 + 0.153/\left(\frac{DV\rho}{\mu}\right)^{0.35}$		LEES [12]
$f = 0.08/\left(\frac{Dv\rho}{\mu}\right)^{1/4}$		WHITE [13]
$f = 0.0054 + 0.006\left(\frac{\mu_c}{DvG_L}\right)$		M'ADAMS AND SHERWOOD [14]

In the above equations:

Q_h = flow rate of gas, std. cu. ft. per hour of 0.6 sp. gr. gas measured at 14.7 psia. and 60° F., at a flowing temperature of 69° F.

G_L = specific gravity (water = 1.00)

μ_c = viscosity, cp.

μ = viscosity, lb./ft.-sec.

All other units previously noted.

where:

M = molecular weight of gas
N = rate of gas flow, weight/unit time
R = gas constant (1,544 ft.-lb./lb.-mole, °F., or equiv.)

Neglecting the log term in the denominator, since it is small in comparison to the first term, yields:

$$Q = K\,(T_s/P_s) \times \left[\frac{(P_1^2 - P_2^2)\,D^5}{fLGT}\right]^{1/2} \quad (9)$$

When written in terms of common pipeline terminology Equation 9 may be written as:

$$Q = (38.77)\,T_s/P_s \times \left[\frac{(P_1^2 - P_2^2)\,D^5}{fLGT_f}\right]^{1/2} \quad (10)$$

where:

T_s = temperature base, °R.
P_s = pressure base, psia.
P_1 = inlet pressure, psia.
P_2 = outlet pressure, psia.
D = internal pipe diameter, in.
L = length of pipe section, miles
G = specific gravity of gas, (air = 1.00)
T_f = gas flowing temperature, °R.
f = friction factor, dimensionless
Q = volume of gas, std. cu. ft./day at T_s and P_s

Friction factor used . . . Most of the common gas flow equations are based on Equation 10 and differ only in the expression used for the friction factor "f." Table 1 summarizes the most familiar equations proposed and the friction factor used. The equations shown are not all that have been proposed but are typical of those that are derived from Equation 10.

These equations may be divided into three general classes:

1. Those based on a constant friction factor.
2. Those where the friction factor is some function of the diameter.
3. Those in which the friction factor is some function of Reynolds modulus ($Dv\rho/\mu$).

Those equations in Class 1 were generally derived for early low-pressure gas systems and have little or no current application. Weymouth's equation and the Panhandle formulas have been most commonly used by the gas industry, although the Oliphant formula[10] is a recent modification that is proposed as being more suitable than the original for high-pressure-diameter lines. The friction factor shown was derived using a typical gas viscosity of 7.4×10^{-6} lb./ft.-sec. If the gas viscosity is substantially different the equation should be multiplied by:

$$(7.4 \times 10^{-6}/\mu)^{0.20}$$

Pipeline Efficiency

Equation 10, the basic flow equation, assumes that the actual line consists of a circular conduit with no restrictions to flow other than pipe-wall friction. In an actual line, however, the presence of liquid, scale, and other foreign substances gives a flow less than that predicted by the equations. In order to correct for this factor it is common to add a multiplying factor E, the pipeline efficiency factor, to all equations.

This factor is a fraction which expresses the actual fraction of the total theoretical flow that will probably flow through a line. It is largely based on experience but Baker and Swerdloff[15] suggest the following for gas-gathering systems:

Type line—	Liquid (gal./M.M.c.f.)	E
Dry-gas field	0.1	0.92
Casing-head gas	7.2	0.77
Gas and condensate	800	0.60
Casing-head gas and crude oil	9,000	0.31
Casing-head gas and crude oil	52,000	0.04

The latter two cases really fall into the range of two-phase flow equations and are probably best solved by them rather than applying a substantial correction to a gas flow equation. In most pipeline applications the factor E ranges from 0.85 to 0.95 in a "clean" line.

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How Compressibility Affects Gas Flow

By Dr. John M. Campbell

AS PRESSURES have risen on natural-gas systems the assumption of ideal gases becomes less valid. It is therefore advisable, and convenient, to incorporate the compressibility factor Z into the flow equations to eliminate this error.

There are several ways in which a compressibility factor may be incorporated into the equations derived under the assumption of ideal gases.

1. Use an average Z at some mean pressure for the section of line studied.
2. Express Z as a function of pressure and temperature.

The use of an average compressibility factor Z_a is convenient, and determined at the average pressure P_a which in turn is found from the equation:

$$P_a = \tfrac{2}{3}\left[\frac{P_1^3 - P_2^3}{P_1^2 - P_2^2}\right] \qquad (1)$$

where P_1 and P_2 are the inlet and outlet absolute pressures respectively. When using this average compressibility factor the basic flow equation (Equation 10, Part 35) is written as:

$$Q = 38.77\,\frac{T_s}{P_s}\left[\frac{(P_1^2 - P_2^2)D^5}{fLG\,T_f Z_a}\right]^{0.5} \qquad (2)$$

The use of an average compressibility factor involves a trial and error solution if one of the pressures is unknown, i.e., where a pressure is to be determined for a given flow rate.

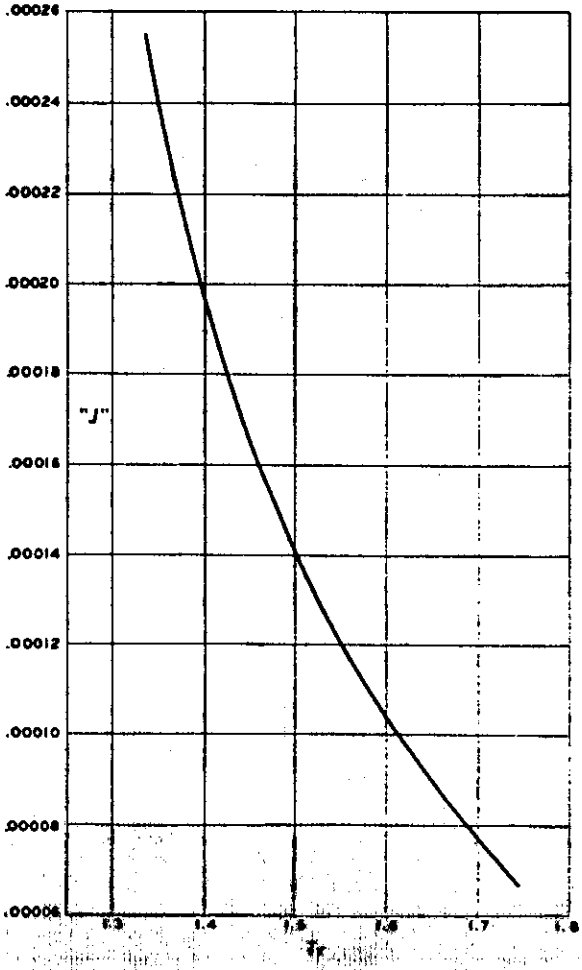
An alternate method is to express Z as a function of pressure with an equation of the form:[1 2 3]

$$\tfrac{1}{2} = Y = C + JP^n \qquad (3)$$

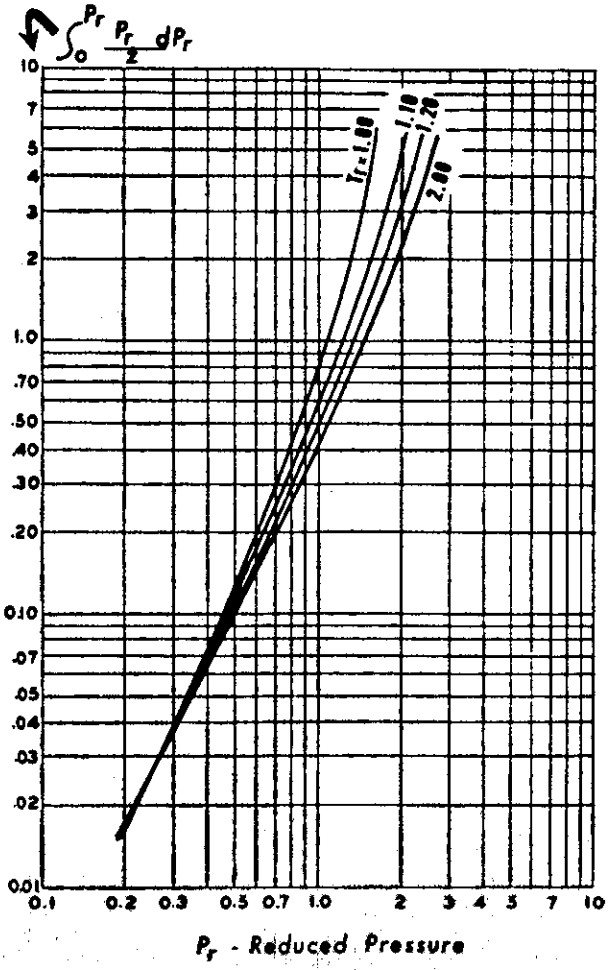
where J is a function of temperature only and n and C are arbitrary constants. In most pipeline applications C and n may be taken as 1.0 so that the equation simplifies to:

$$Y = 1/Z = 1 + JP \qquad (4)$$

The factor J may be easily handled as long as the flowing temperature is substantially constant, an assumption



VARIATION OF J-FACTOR with reduced temperature, T_r. Fig. 1.



INTEGRAL OF $(P_r/Z)\,dP_r$ as a function of T_r and P_r. Fig. 2.

that is generally satisfactory as long as the ground temperature does not change markedly. If the above expression is introduced into the differential form of the gas flow equation the basic flow equation may be written:

$$Q = 38.77 \frac{T_s}{P_s}\left[\frac{P_1^2(1+\frac{2}{3}JP_1) - P_2^2(1+\frac{2}{3}JP_2)}{fLGT_f}\right]^{1/2} D^{5/2} \quad (5)$$

The value of J for use in this expression may be found from Fig. 1.[3] The value of reduced temperature T_r for use on the abscissa is found by one of the methods discussed in Part 33.

The above procedure of assuming an ideal gas and then correcting the result by empirical means is questionable although the final result is very applicable. Clinedinst[4] has therefore derived an equation where the Z is incorporated throughout, the final result being:

$$Q = 54.83 \frac{T_s}{P_s}\left[\frac{\int_{P_2}^{P_1} \frac{P}{Z}\,dP}{fGTL}\right]^{1/2} D^{5/2} \quad (6)$$

when the equation is expressed in terms of the friction factor defined by Equation 6, Part 35.

This equation is equivalent to Equations 2 and 5 and may be used with any appropriate friction factor. The author,[4] however, proposes the correlation:

$$\log_{10}(4f) = -1.7447 - 0.010637D \quad (7)$$

The integral $\int (P/Z)\,dP$ obviously offers difficulty since Z is some function of P, which prevents direct integration. Graphical integration is possible but it is tedious and time consuming. Consequently several alternative methods of evaluating this integral have been developed.[1 4 5]

The integral may be solved by substituting the value of Z in Equation 4. into the integral, or:

$$\int_{P_2}^{P_1} \frac{P}{Z}\,dP = \int_{P_2}^{P_1} (P + JP^2) = \frac{J}{3}(P_1^3 - P_2^3) + \frac{1}{2}(P_1^2 - P_2^2) \quad (8)$$

A second solution is possible by expressing the integral in terms of reduced pressure, P_r.

$$\int_{P_2}^{P_1} \frac{P}{Z}\,dP = P_c^2 \int_{Pr_2}^{Pr_1} \frac{P_r}{Z}\,dP_r = P_c^2\left[\int_0^{Pr_1} \frac{P_r}{Z}\,dP_r - \int_0^{Pr_2} \frac{P_r}{Z}\,dP_r\right] \quad (9)$$

where:

P_c = critical or pseudocritical pressure

P_r = reduced pressure = P/P_c

Equation 9 may be conveniently solved by preparing a plot of the type shown in Fig. 2. The general application and accuracy of such a correlation are limited by the "Z" chart from which it is derived and the scale used for the data. However, the steepness of the curve in Fig. 2 introduces potential error, for a small error in P_r will give a magnified error in factor J.

A similar approach solves the integral by the use of "F" factors which are obtained from a generalized chart.[5]

Example problem . . . Determine the value of the integral,

$$\int_{P_2}^{P_1} \frac{P}{Z}\,dP$$

by two different methods for a natural gas having a critical temperature of 360° R. and a critical pressure of 672 psia., if the average flowing temperature is 65° F., the initial pressure is 946 psia. and the final pressure 638 psia.

Solution 1 (Equation 8):

$$T_r = \frac{460 + 65}{360} = 1.46$$

∴ From Fig. 1, J = 0.000159

Solution 1 (*Equation 8*)

$$\int_{P_2}^{P_1} \frac{P}{Z}\,dP = \frac{0.000159}{3}\left[(946)^3 - (638)^3\right] + \frac{1}{2}\left[(946)^2 - (638)^2\right]$$

$$= \underline{2.71 \times 10^5}$$

Solution 2 (*Equation 9*)

$$P_{r_1} = \frac{946}{672} = 1.41, \quad P_{r_2} = \frac{638}{672} = 0.95$$

From Fig. 2, $\int_0^{P_r} \frac{P_r}{Z}\,dP_r = 1.0 \text{ @ } P_{r_1} = 1.41$

$= 0.42 \text{ @ } P_{r_2} = 0.95$

$$\int_{P_2}^{P_1} \frac{P}{Z}\,dP = (672)^2(1 - 0.42) = \underline{2.61 \times 10^5}$$

The above difference is less than 4 per cent which is equivalent to the potential error as a result of using a slide rule on the above calculations.

Other methods . . . Several other convenient equations have been proposed for the solution of flow problems.[6 7] The equation of Poettman[6] is similar in form to Equations 6 and 9, except that the reduced integral is expressed as $(Z/P_r)\,dP_r$ and is integrated from 0.2 to P_r rather than 0 to P_r. Values of this integral are shown in Part 39 under the discussion of calculation of bottom-hole pressure. Baker and Swerdloff[7] have presented a basic solution of Equation 2 where the terms are conveniently consolidated in graphical form for easy solution.

It is obvious from the foregoing discussion that the designer of gas lines has a wide choice of equations. They differ primarily with the friction factor and the method of handling the compressibility factor. Most major pipelines, however, use some form of the Panhandle or Weymouth equations, as dictated by company policy. In the design of gathering systems the exact form used will not only depend on company policy, but on the inevitable personal preference of the individual engineer.

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How to Calculate Complex-System Gas Flow

By Dr. John M. Campbell

MANY gas-flow calculations involve some form of complex system. When such a need arises it becomes necessary to properly analyze the system before making the necessary flow calculations. In fact, the analytical portion of the problem is all important, for any of the applicable flow equations may be used.

For convenience, all derivations are made with Weymouth's equation, but Table 1 shows the equivalent form of the Panhandle and Clinedinst equations.

Systems of Parallel Lines

Parallel or loop lines are used to increase the amount of flow per pound of pressure drop. In one case a line is looped to increase the capacity without increasing the pressure drop. In another, the pressure on the line must be lowered, so a loop is added to maintain the desired flow rate.

Fig. 1a illustrates a simple loop system where original line A has been partially looped with line B. The purpose of line B is to decrease the pressure drop between P_1 and P_A per unit quantity of gas flowing, so that the reduction will compensate for the increased pressure drop from P_A to P_2, when the flow rate is increased, or the pressure is lowered.

Using the same principles as those applied to liquid flow in Part 31, it is apparent from Fig. 1a that

$$Q_A + Q_B = Q_C = Q\ \text{Total} \quad (1)$$

$$\Delta P_A = \Delta P_B \text{ (since they have two points in common)} \quad (2)$$

$$\Delta P_A\ (\text{or}\ \Delta P_B) + \Delta P_C = \Delta P\ \text{Total} \quad (3)$$

Now Weymouth's equation may be written as

$$Q = 433.45\ T_s/P_s \times [(P_1^2 - P_2^2)/LGTZ_a]^{1/2}\ D^{8/3} \quad (4)$$

and may be applied for any single line between any two points between which no work is added or withdrawn.

Inasmuch as the same gas is flowing through both lines, Equation 4 may be written

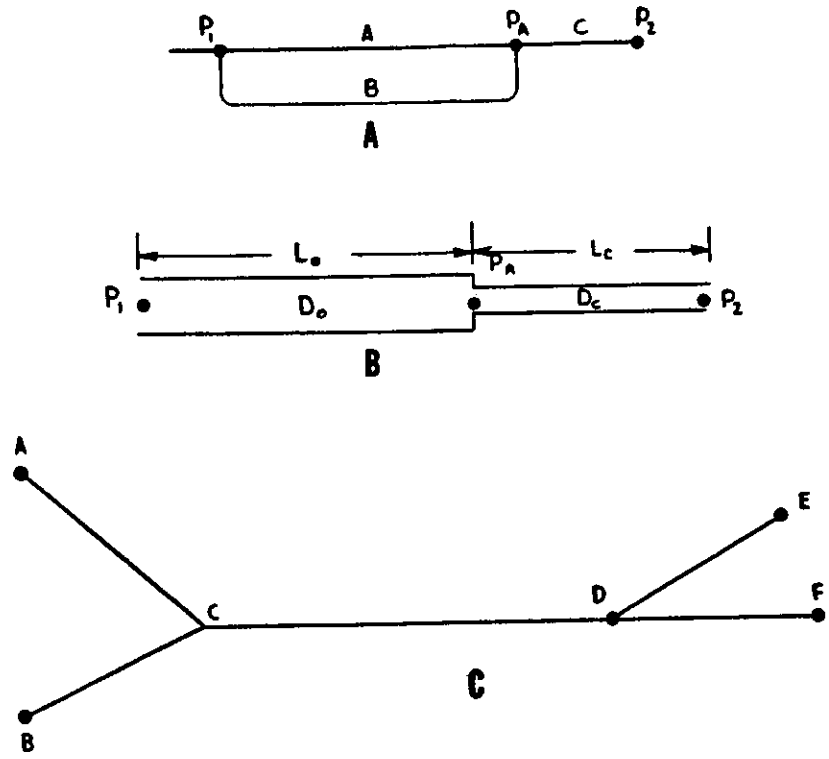
$$Q = K\ (D^{16/3}/L)^{1/2} \quad (5)$$

where K encompasses all of the terms that are constant among the loop lines.

Table 1—Comparison of Complex Formulas for Different Flow Equations

QUANTITY	WEYMOUTH	CLINEDINST	PANHANDLE
LINES IN SERIES			
EQUIVALENT DIAMETER	$D_1 = D_2 \left(\frac{L_1}{L_2}\right)^{3/16}$	$D_1 = D_2 \left(\frac{L_1}{L_2}\right)^{1/5}$	$D_1 = D_2 \left(\frac{L_1}{L_2}\right)^{0.2060}$
EQUIVALENT LENGTH	$L_1 = L_2 \left(\frac{D_1}{D_2}\right)^{16/3}$	$L_1 = L_2 \left(\frac{D_1}{D_2}\right)^{5}$	$L_1 = L_2 \left(\frac{D_1}{D_2}\right)^{4.854}$
LINES IN PARALLEL			
EQUIVALENT DIAMETER	$\frac{D_o^{8/3}}{L_o^{1/2}} = \frac{D_1^{8/3}}{L_1^{1/2}} + \frac{D_2^{8/3}}{L_2^{1/2}}$	$\left(\frac{D_o^5}{L_o}\right)^{1/2} = \left(\frac{D_1^5}{L_1}\right)^{1/2} + \left(\frac{D_2^5}{L_2}\right)^{1/2}$	$\frac{D_o^{2.618}}{L_o^{0.5394}} = \frac{D_1^{2.618}}{L_1^{0.5394}} + \frac{D_2^{2.618}}{L_2^{0.5394}}$
LOOPING REQUIREMENTS*	$X = \frac{1-\left(\frac{Q}{Q_1}\right)^2}{1-\left[\frac{D^{8/3}}{D^{8/3}+D_1^{8/3}}\right]^2}$	$X = \frac{1-\left(\frac{Q}{Q_1}\right)^2}{1-\left[\frac{D^{5/2}}{D^{5/2}+D_1^{5/2}}\right]^2}$	
ENTIRE LINE LOOPED*	$\frac{Q_1}{Q} = \left[1+\left(\frac{D_1}{D}\right)^{8/3}\right]$	$\frac{Q_1}{Q} = \left[1+\left(\frac{D_1}{D}\right)^{5/2}\right]$	
DIAMETER OF ORIGINAL AND PARALLEL LINES THE SAME*	$X = 4/3\left[1-\left(\frac{Q}{Q_1}\right)^2\right]$	$X = 4/3\left[1-\left(\frac{Q}{Q_1}\right)^2\right]$	

*ASSUMES THAT THE LENGTH OF ALL LINES IN THE LOOP SECTION ARE THE SAME LENGTH



A SIMPLE LOOP SYSTEM is shown (A), the equivalent single line (B), and a branch system (C). Fig. 1.

Equivalent of loop . . . The flow characteristics of loop lines A and B may be expressed in terms of an equivalent single line of diameter D_o and L_o. This equivalent line is defined as a single line that has the same capacity as the loop system when the same pressure drop is imposed. If Equation 5 is written for each line and substituted into Equation 1, the K terms cancel and:

$$\frac{D_o^{8/3}}{L_o^{1/2}} = \frac{D_A^{8/3}}{L_A^{1/2}} + \frac{D_B^{8/3}}{L_B^{1/2}} \quad (6)$$

If the lines are the same length, the denominators cancel. The loop system in Fig. 1a then reduces to the system shown in Fig. 1b. With reference to the latter figure,

$$Q_o = Q_C \text{ and } (\Delta P/L)_o\,(L_o) + (\Delta P/L)_c\,(L_c) = \text{Total } \Delta P \quad (7)$$

This system could be solved, in the absence of knowing the pressure at P_A, by writing an equation for each section, equating them, and solving for P_A. With this value either of the equations may then be solved for the unknown quantity.

Equivalent of series . . . However, it is more convenient to express this system in terms of a single line that is equivalent to the two lines in series. This is done by finding what length of line D_o would be equivalent to L_c miles of diameter D_c, or vice versa. In this case equivalent means that both lines would have the same capacity with the same pressure drop. In general terms this may be found by writing Equation 5 for each line and substituting them into Equation 7. Then:

$$(D_1^{16/3}/L_1)^{1/2} = (D_2^{16/3}/L_2)^{1/2} \quad (8)$$

and squaring both sides:

$$L_1 = L_2\,(D_1/D_2)^{16/3}$$
$$\text{or } D_1 = D_2\,(L_1/L_2)^{3/16} \quad (9)$$

where L_1 = length of line of diameter D_1 that is equivalent to a line of diameter D_2 and length L_2.

Example: If in Fig. 1b, D_o = 6 in., L_o = 10 miles, D_c = 8 in., and L_c = 20 miles, determine the proper length and diameter of a single line that would be equivalent to the system shown.

Solution 1:

$L_1 = 20\,(6/8)^{16/3} =$ **4.3 miles**

Total length = 10 + 4.3
= **14.3 miles of 6-in.**

Solution 2:

$L_1 = 10\,(8/6)^{16/3} =$ **46.2 miles**

Total length = 20 + 46.2
= **66.2 miles of 8-in.**

This solution says 14.3 miles of 6-in. or 66.8 miles of 8-in. represents a single-line system that is equivalent to that shown in Fig. 1b. Consequently, the substitution of either combination of values into Equation 4 yields a solution for the loop system.

The solution of the branch line system shown in Fig. 1c must be basically solved by remembering that

$$Q_{BC} + Q_{AC} = Q_{CD} = Q_{DE} + Q_{DF} \quad (10)$$

It is usually necessary to substitute the appropriate flow equation into this expression and solve for the unknown quantity. An example of this type of solution is shown in the Part 38, to follow.

Nomenclature

For lines in series:

D_1 = diameter of line of length L_1 that is equivalent to a line of length L_2 and diameter D_2
L_1 = length of line of diameter D_1 that is equivalent to a line of length L_2 and diameter D_2

For lines in parallel:

D_o = diameter of a single line that is equivalent to a group of parallel lines
D = diameter of the original line before looping
D_1 = diameter of a single loop line (or equivalent diameter of a group of loop lines)
D_2 = diameter of a second loop line in parallel with line D_1
L_o = length of equivalent single line corresponding to D_o
L_1 = length of loop line D_1
L_2 = length of loop line D_2
Q = original flow rate before line is looped
Q_1 = flow rate after looping line
X = fraction of length of original line that is looped

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ELEMENTS OF FIELD PROCESSING—38

Examples Showing Application Of Gas-Flow Equations

By Dr. John M. Campbell

THE FOLLOWING examples will serve to illustrate the application of the gas flow equations developed in Parts 34-37. For simplicity, the Weymouth equation, uncorrected for gas compressibility, will be used as an illustration of the principles involved.

Example 1 . . . A portion of a large, gas-gathering system consists of a 6.067-in. i.d. line 9.4 miles long that is handling 7.6 M.M.c.f.d. of gas with an average specific gravity of 0.64. The pressure at the upstream end of this section is 375 psig., and the average delivery pressure is 300 psig. The average temperature is 73° F. Due to new well completions, it is desired to increase the capacity of this line 20 per cent by looping with additional 6.067-in. i.d. line. What length is required?

Solution . . . Let Y represent the length of the loop section as shown below:



The new flow rate $Q_1 = (1.2)(7.6) = 9.1$ **M.M.c.f.d.**

The looped section may be represented by a single line D_o. From Equation 6 (Part 37), remembering that all lengths are the same:

$$D_o = [(6.07)^{5/2} + (6.07)^{5/2}]^{2/5}$$

$$= 7.9 \text{ in. I.d.}$$

A Weymouth equation for each section may now be written:

$$9.1 \times 10^6 = (433.5)(520/14.65) \times \left[\frac{(390)^2 - P_A^2}{(Y)(0.64)(533)}\right]^{1/2} (7.9)^{8/3}$$

$$9.1 \times 10^6 = (433.5)(520/14.65) \times \left[\frac{P_A^2 - (315)^2}{(9.4-Y)(0.64)(533)}\right]^{1/2} (6.07)^{8/3}$$

Note that 14.7 psi. has been added to each gage pressure to convert it to absolute pressure. Since there are two independent equations and only two unknowns, these questions may be solved algebraically for Y.

Each equation may be solved for P_A and Y to yield:

$$7{,}950Y + P_A^2 = 175{,}000$$

$$1{,}948Y + P_A^2 = 152{,}000$$

If one is subtracted from the other to eliminate P_A:

$$6{,}002Y = 23{,}000 \text{ or } Y = 3.83 \text{ miles}$$

This is then the necessary length of the loop (parallel) line to increase the flow. The same approach would be used if it were desired to lower the line pressure at the same flow rate.

The above represents the general algebraic solution of the problem. In this case, the equation for "Looping Requirements" shown in Table 1 of Part 37 may be used and represents an easier solution.

$$X = \frac{1 - (Q/Q_1)^2}{1 - [D^{5/3}/(D^{5/3} + D_1^{5/3})]^2}$$

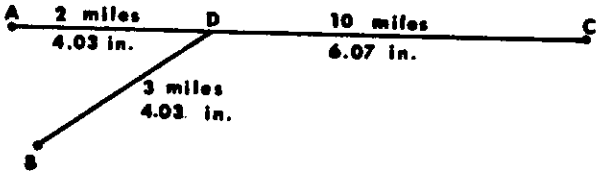
$$= \frac{1 - (7.6/9.1)^2}{1 - \left[\frac{(6.07)^{5/3}}{(6.07)^{5/3} + (6.07)^{5/3}}\right]^2}$$

$$= 0.407$$

Therefore, the length of the loop section is equal to:

$$(0.407)(9.4) = 3.82 \text{ miles}$$

Example 2 . . . The following system is to be designed for the gathering of gas. Well "A" is a flowing gas well that will deliver 3.0 M.M.c.f.d. Lease "B" contains high-gas-oil-ratio oil wells. It is necessary to determine what compressor discharge pressure is necessary in order to deliver 1.75 M.M.c.f.d. from Point B. The pressure at the transmission line (Point C) is 300 psig. Average flowing temperature will be 85° F. The gravity of the flowing gas is 0.63, and that of the casing-head gas is 0.71. The line diameters shown have been arbitrarily fixed because of pipe availability.



It is obvious that once the pressure at Point D is obtained, P_B may be found. Furthermore:

$$Q_{AD} + Q_{BD} = Q_{CD}$$

The average specific gravity in line CD is estimated to be:

$$\frac{(3.0)(0.63) + (1.75)(0.71)}{4.75} = 0.66$$

For line CD:

$$4.75 \times 10^6 = 433.5\,(520/14.65) \times \left[\frac{P_D^2 - (315)^2}{(10)(0.66)(545)}\right]^{1/2} (6.07)^{8/3}$$

$$\text{or } P_D = 350 \text{ psia.}$$

Solving for line BD:

$$1.75 \times 10^6 = (433.5)(520/545) \times \left[\frac{P_B^2 - (350)^2}{(3)(0.71)(545)}\right]^{1/2} (4.03)^{8/3}$$

$$P_B = 362 \text{ psia.} = 347 \text{ psig.}$$

This, therefore, is the required compressor discharge pressure if the compressibility factor is neglected. If the compressibility had been incorporated, as shown in previous discussions, the pressure required would have been a few per cent less.

In actual practice it would be desirable to incorporate a pipeline efficiency factor into the solution if any quantity of liquids was anticipated (Part 35). In the pressure range studied, with an efficiency of about 90 per cent, the pressure would be increased about 5 psi. In any event, a 10 to 20-psi. safety factor should be incorporated into the discharge pressure to allow for unforeseen factors that may arise. **End.**

Simple Method Determines Gas Flow Rate

by William C. Barron, *Production Foreman, Marathon Oil Co., Anchorage, Alaska*

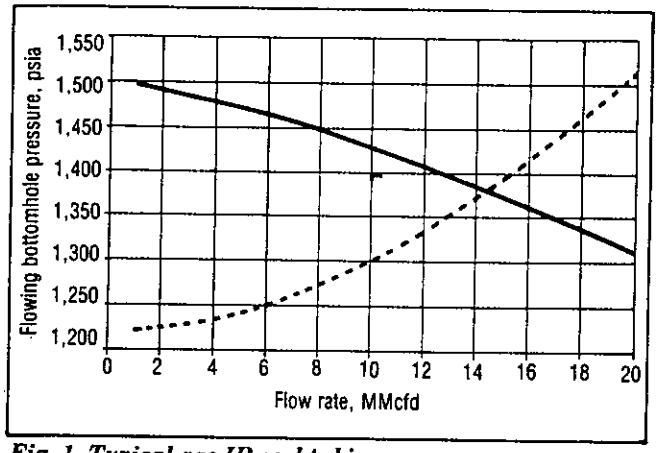


Fig. 1. Typical gas IP and tubing curve.

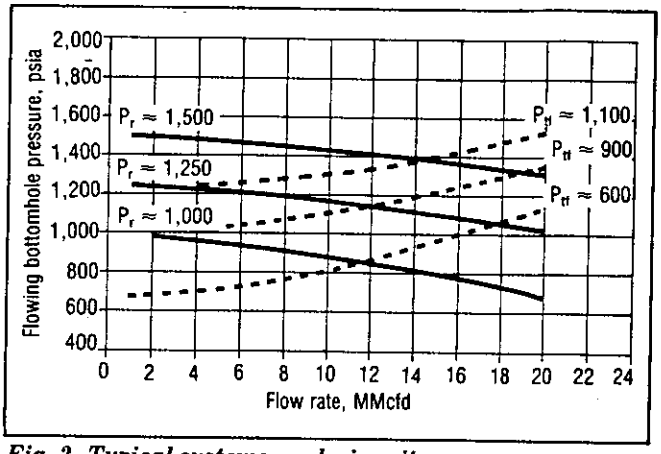


Fig. 2. Typical systems analysis suite.

A simple analytical method has been developed that accelerates the determination of theoretical surface flow rates for shallow dry gas wells (less than 8,000 ft). For several years, the algorithm has proven accurate for on-site well testing, well design analysis, and deliverability projections at widely varying static reservoir and system operating pressures.

By directly calculating the flowing bottomhole pressure and the theoretical surface flow rate, the technique circumvents the graphic interpolation of inflow performance and tubing curves. Using the classic reservoir backpressure equation and a modified Cullender and Smith vertical flow correlation, the algorithm can be easily programmed on a programmable calculator or a microcomputer.

Systems Analysis

Systems analysis is the evaluation of pressure drop and flow potential through the various subsystems of a producing well. The two major subsystems are the reservoir and well bore. Reservoir inflow performance (IP) for an individual gas well can be described by the classic radial Darcy's Law equation or one of its many enhancements.[1-7] These inflow relationships are used to develop a series of curves estimating reservoir flow potential at varying static reservoir and flowing bottomhole pressures (solid line in Fig. 1). Pressure drop through the well bore subsystem is described by numerous correlations derived from the general energy equation[2, 8-10]. These correlations represent total pressure drop and flow potential through vertical and inclined tubulars as a series of tubing curves (dash line in Fig. 1).

The intersection of an IP and tubing curve defines the well's theoretical surface flow rate and is the only common point between the two subsystems. At a specified flow rate, the pressure drop through the reservoir and the tubing yield an equivalent flowing bottomhole pressure. An energy balance exists between the two subsystems. A higher flow rate requires a greater flowing bottomhole pressure than the reservoir can yield – a reservoir energy deficit. At lower flow rates, the opposite conditions exist – an energy surplus.

The combination, or series, of curves from each subsystem generates a suite of curves which identifies the theoretical surface flow rate at various reservoir and surface pressures (Fig. 2). This suite of curves is a valuable tool for projecting well deliverability and surface facility operating conditions through the life of the well. However, this graphic representation is time consuming to create and its accuracy limited by required interpolation.

Analytical Alternative

This presentation uses the classic reservoir backpressure equation, of the form:

$$Q_r = C(P_{ws}^2 - P_{wf}^2)^n \qquad (1)$$

This correlation describes a reservoir's flow potential as a straight line when presented in terms of $P_{ws}^2 - P_{wf}^2$ versus Q_r on log-log coordinates (Fig. 3). The variables C and n are unique to each well. The coefficient C is a function of reservoir, completion, and gas properties. Variance from radial Darcy Law, caused by flow turbulence, is expressed by the n exponent (that is, $n \neq 1.0$). Although the standard back-

of the surface and bottomhole values. This is the basis of the average temperature and compressibility correlation. The greatest area of probable error in this assumption is the determination of the compressibility terms. The ability to define gas compressibility becomes important in conditions where major changes in pressure and temperature occur. This algorithm uses the Dranchuk. Purvis. and Robinson correlation.[11]

The effects of kinetic energy are generally an insignificant portion of the total pressure drop through the well bore; therefore, a common simplification of vertical flow correlations is to neglect this pressure drop. One exception to this assumption is in low-pressure environments where total pressure drop in high velocity flow may be sufficient to make the kinetic energy term significant.

Finally, the information required to evaluate the reservoir subsystem can be obtained through various means. Regardless of the means, regression analysis techniques should be used to determine the variable terms of the reservoir backpressure equation. Obviously, errors in their determination compound subsequent error from their use.

In conclusion, here are several points to consider:

- The algorithm presented, programmed on either a programmable calculator (Table 2) or microcomputer, has proven to be reliable and accurate in determining the flowing bottomhole pressure and surface flow rate of shallow gas wells.
- By eliminating graphic interpolation required with standard system curves, use of the algorithm can increase accuracy of systems analysis. The algorithm also has the flexibility to be used during on-site well testing, as well as for long-term well deliverability projections.
- The derivation allows for the use of a variety of inflow performance relationships, other than the classic backpressure equation. An equation defining reservoir inflow performance simplified to the form, $Q_r = C(P_{ws}^2 - P_{wf}^2)^n$, can be readily used in place of the back pressure equation.

Nomenclature

a = absolute roughness, in.
C = numerical flow coefficient, MMcfd/psi^2
D_i = internal pipe diameter, in.
f = Fanning coefficient of friction
f_c = coefficient of friction for Colebrook equation
f_g = initial determination of Fanning coefficient of friction
F_r = dimensionaless numerical friction coefficient
F_* = Fanning coefficient of friction of Nikuradse and von Karman
g = gas gravity (air = 1.0)
H = well bore true vertical depth, ft
L = well bore measured depth, ft
n = dimensionaless numerical flow exponent
N_{re} = Reynolds number
P_{tf} = flowing surface pressure, psi
P_{wf} = flowing bottomhole pressure, psi
P_{ws} = static reservoir pressure, psi
Q_r = reservoir flow rate, MMcfd
Q_s = surface flow rate, MMcfd
P_r = reservoir pressure, psi
s = dimensionaless numerical flow coefficient
T = average well bore temperature, °R
u = gas viscosity, cp
Z = average gas compressibility term

Acknowledgement. The author would like to thank Marathon Oil Co. for permission to publish this paper and the management and professional staff of the Alaskan District of Marathon Oil Co., in particular D. DeYoung. Special thanks go to T. and V. Arnold.

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Appendix

The integral form of the Cullender and Smith correlation is given as follows:[8]

$$\frac{1000GL}{53.33} = \int_{P_{wf}}^{P_{tf}} \frac{P/TZ\ d(P)}{F^2 + (H/1000\ L)\ (P/TZ)^2} \qquad (A1)$$

where

$$F^2 = 2.6665\ fQ_s^2/D_i \qquad (A2)$$

This correlation, as published, assumed all turbulent flow. An equation defining the Fanning friction factor (f) as a function of absolute roughness and pipe diameter was derived as follows:[8]

$$f = (4.372\ X\ 10\text{-}3)/D_i^{0.223} \qquad (A3)$$

This approximation of the Fanning friction factor is for pipe diameters less than 4.277 in. with an absolute roughness of 0.0006 in. A more rigorous initial calculation of the friction variable for rough pipe is based upon the work of Nikuradse and von Karman.[10]

$$F_* = 1/(9.2103\ \ln(3.7D_i \div a)^2 \qquad (A4)$$

If the flow is not completely turbulent, potential error can be minimized through use of the universal Colebrook equation to determine a more accurate friction factor.[4,9]

$$f_c = [(1/(1.74 - (2\ \ln[(2a/D_i) + (18.7(/N_{re}f_g^{0.5}))]))^2]/4 \qquad (A5)$$

where

$$N_{re} = 20010\ gQ/(uD_i) \qquad (A6)$$

The Colebrook equation was derived to describe the friction factor as a function of the Reynolds number in terms of pipe roughness, diameter, and flow rate. This equation is very useful in determining friction factors in the transition flow region of the Moody Diagram.

About the Author



William C. Barron is a production foreman for Marathon Oil Co. currently working in the McArthur River field in Anchorage, Alaska. He is a graduate of the University of Texas at Austin with a degree in petroleum engineering. He is a member of the Society of Petroleum Engineers.

■

TABLE 2. Algorithm program listing for HP-41CY

Hewlett-Packard Petroleum Pack required.

Size: 055; User mode; "R/S" after each variable input

Labels: A: Input of variables
B: Initiate calculation
C: Review of variables and calculated values

"Est % DRWDN" = $((P_{ws}-P_{wf})/P_{ws})$ 100 is used as initial "guess" for Newton-Raphson interaction.

Note: Recalculation of same problem with minor variable change is accomplished by keying "R/S" for those variables which are to remain constant.

```
01LBL "GROP"
02 SF 12
03 "GROP"
04 AVIEW
05 CF 12
06 "WCBARRON"
07 AVIEW
08 ADV
09 "A=INPUT"
10 " B=CALC"
11 " C=REVIEW"
12 AVIEW
13 CF 05
14 RTN
15LBL A
16 XEQ "INPUT"
17 RTN
18LBL B
19 FIX 2
20 XEQ "LOOP"
21 XEQ "FLOW"
22 RCL 07
23 STO 41
24 XEQ "FRAB"
25 RCL 06
26 STO 45
27 TONE 0
28 XEQ "OUTPUT"
29 RTN
30LBL C
31 XEQ "REVIEW"
32 RTN
33LBL "INPUT"
34 RCL 20
35 "C COEFF"
36 PROMPT
37 STO 20
38 RCL 23
39 "N EXPONENT"
40 PROMPT
41 STO 23
42 RCL 24
43 "RES PRESS"
44 PROMPT
45 STO 24
46 RCL 25
47 459.67
48 -
49 "RES TEMP, F"
50 PROMPT
51 459.67
52 +
53 STO 25
54 0.0
55 STO 26
56 STO 27
57 STO 28
58 RCL 15
59 "GAS GRAVITY"
60 PROMPT
61 STO 15
62 XROM "CTcPc"
63 STO 11
64 X<>Y
65 STO 10
66 RCL 30
67 "TUBING ID"
68 PROMPT
69 STO 30
70 RCL 31
71 "MEAS DEPTH"
72 PROMPT
73 STO 31
74 RCL 32
75 "TRUE VERT"
76 PROMPT
77 STO 32
78 RCL 33
79 "ABS ROUGH"
80 PROMPT
81 STO 33
82 RCL 34
83 "FLOW PRES"
84 PROMPT
85 STO 34
86 RCL 35
87 459.67
88 -
89 "FLOW TEMP, F"
90 PROMPT
91 459.67
92 +
93 STO 35
94 RCL 49
95 "EST %DRWDN"
96 PROMPT
97 100
98 /
99 CHS
100 1
101 +
102 X^2
103 CHS
104 1
105 +
106 STO 49
107 ADV
108 RTN
109LBL "REVIEW"
110 FIX 6
111 "C COEFF "
112 ARCL 20
113 AVIEW
114 FIX 3
115 "N EXP "
116 ARCL 23
117 AVIEW
118 FIX 1
119 "SIBHP "
120 ARCL 24
121 AVIEW
122 RCL 29
123 459.67
124 -
125 STO 06
126 "BHT, F "
127 ARCL 06
128 AVIEW
129 FIX 3
130 "GAS GRAV "
131 ARCL 15
132 AVIEW
133 "RESI "
134 ARCL 36
135 AVIEW
136 "RESVIS "
137 ARCL 37
138 AVIEW
139 ADV
140 "TBG ID "
141 ARCL 30
142 AVIEW
143 FIX 0
144 "MD "
145 ARCL 31
146 AVIEW
147 "TVD "
148 ARCL 32
149 AVIEW
150 FIX 5
151 "ABS ROUGH "
152 ARCL 33
153 AVIEW
154 FIX 1
155 "FTP "
156 ARCL 34
157 AVIEW
158 RCL 35
159 459.67
160 -
161 STO 06
162 "FTT, F "
163 ARCL X
164 AVIEW
165 FIX 3
166 "SURFZ "
167 ARCL 38
168 AVIEW
169 "SURFVIS "
170 ARCL 39
171 AVIEW
172 ADV
173 ADV
174 ADV
175 ADV
176 ADV
177 RTN
178LBL "LOOP"
179 RCL 29
180 459.67
181 -
182 STO 16
183 LASTX
184 +
185 RCL 10
186 /
187 STO 18
188 RCL 24
189 STO 17
190 RCL 11
191 /
192 STO 15
193 RCL 16
194 X<>Y
195 CZ
196 STO 36
197 RDN
198 LASTX
199 XROM "CU6"
200 STO 37
201 RCL 35
202 459.67
203 -
204 STO 16
205 LASTX
206 +
207 RCL 10
208 /
209 STO 16
210 RCL 34
211 STO 17
212 RCL 11
213 /
214 STO 19
215 RCL 16
216 X<>Y
217 CZ
218 STO 38
219 RDN
220 LASTX
221 XROM "CU6"
222 STO 39
223 RCL 37
224 +
225 2
226 /
227 STO 09
228 RCL 36
229 RCL 38
230 +
231 2
232 /
233 STO 08
234 RCL 29
235 RCL 35
236 +
237 2
238 /
239 STO 12
240 RCL 15
241 RCL 32
242 *
243 0.03784
244 *
245 RCL 12
246 /
247 RCL 08
248 /
249 STO 40
250 RCL 30
251 RCL 33
252 /
253 3.7
254 *
255 LOG
256 4
257 *
258 X^2
259 1/X
260 STO 41
261 RCL 24
262 X^2
263 RCL 49
264 *
265 STO 45
266LBL "FRAB"
267 RCL 41
268 RCL 30
269 5
270 Y^X
271 /
272 2.6665
273 *
274 SQRT
275 STO 42
276 RCL 12
277 *
278 RCL 08
279 *
280 RCL 20
281 *
282 1/X
283 X^2
284 STO 06
285 RCL 40
286 E^X
287 1
288 -
289 RCL 31
290 *
291 1/X
292 RCL 32
293 *
294 RCL 06
295 *
296 0.001
297 *
298 STO 43
299 RCL 40
300 E^X
301 RCL 34
302 X^2
303 +
304 STO 44
305 100.11001
306 STO 26
307LBL "YFI"
308 RCL 45
309 RCL 23
310 2
311 *
312 Y^X
313 RCL 45
314 RCL 43
315 *
316 +
317 RCL 24
318 X^2
319 CHS
320 RCL 44
321 +
322 RCL 43
323 *
324 +
325 STO 50
326 RCL 45
327 RCL 23
328 2
329 *
330 1
331 -
332 Y^X
333 RCL 23
334 *
335 2
336 *
337 RCL 43
338 +
339 STO 51
340 RCL 50
341 X<>Y
342 /
343 CHS
344 RCL 45
345 +
346 STO 06
347 RCL 45
348 -
349 ABS
350 10.0
351 X>Y?
352 RTN
353LBL "COUNT"
354 TONE 9
355 ISG 26
356 GTO 01
357 "ITERATION LIMIT"
358 AVIEW
359 GTO "GROP"
360 RTN
361LBL 01
362 RCL 06
363 ABS
364 STO 45
365 GTO "YFI"
366LBL "FLOW"
367 RCL 06
368 STO 45
369 RCL 23
370 Y^X
371 RCL 20
372 *
373 STO 46
374LBL "Re"
375 20010
376 *
377 RCL 15
378 *
379 RCL 09
380 /
381 RCL 30
382 /
383 STO 47
384LBL "CLBRK"
385 RCL 41
386 SQRT
387 RCL 47
388 *
389 1/X
390 18.7
391 *
392 2
393 RCL 33
394 *
395 RCL 30
396 /
397 +
398 LOG
399 2
400 *
401 CHS
402 1.74
403 +
404 1/X
405 X^2
406 4
407 /
408 STO 07
409 RCL 41
410 -
411 ABS
412 0.0001
413 X>Y?
414 RTN
415 RCL 07
416 STO 41
417 GTO "CLBRK"
418 RTN
419LBL "OUTPUT"
420 RCL 23
421 Y^X
422 RCL 20
423 *
424 STO 46
425 "Q= "
426 ARCL 46
427 " MMCF/D"
428 AVIEW
429 PSE
430 "FTP= "
431 ARCL 34
432 " PSI"
433 AVIEW
434 PSE
435 RCL 24
436 X^2
437 RCL 45
438 -
439 0.5
440 Y^X
441 STO 48
442 "FBHP= "
443 ARCL 48
444 " PSI"
445 AVIEW
446 PSE
447 "BHP= "
448 ARCL 24
449 " PSI"
450 AVIEW
451 PSE
452 RCL 24
453 RCL 48
454 -
455 RCL 24
456 /
457 100
458 *
459 STO 49
460 "%DRW DN= "
461 ARCL X
462 AVIEW
463 ADV
464 ADV
465 ADV
466 ADV
467 ADV
468 .END.
```